Managing energy efficiently in hotels and commercial buildings

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Editors

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ed building sector, which is also widely diffused
nt to change, has become a major offender
e use of electricity and petroleum products.
ization, with the resultant demand for housing
buildings, will put more pressure on energy
ate energy costs. Electricity, which has the
ate of all forms of energy used in the country, is
supply in many parts of the country. The short-
and capacity is particularly acute. Huge invest-
ed to provide fresh generation capacity. The
ironmental degradation and pollution will
ituation further.

ial increase in the costs of different forms of
the growing importance of energy conservation
tting energy costs in buildings. Studies have
orporation of energy-efficiency concepts reduces
w buildings by 40% of that in conventional
existing buildings by 20% by suitable retrofit
wever, lack of awareness and non-availability of
s for decision-making discourage many from
-efficiency concepts in their buildings.

ion, titled *Managing energy efficiently in hotels and*
ngs, provides a set of guidelines and methodolo-
out energy conservation studies, popularly
audits in buildings. Apart from outlining some
on energy conservation in hotels and commer-
is book focuses on almost all energy-consuming
engineering and maintenance professionals to
e audits. TERI's expertise and experience in the
onservation in hotels and commercial buildings
use in developing the contents of this book.

efficiently in hotels and commercial buildings is
ted at engineering and maintenance profession-
neers, system designers, academics, and consult-
this field.

R K Pachauri
Director-General, TERI

nt of housing and commercial buildings
ɛ pressure on our dwindling energy
ity of energy is limited and known
re exhausting fast. Therefore, energy
; importance in different sectors. In view
:risis and increasing energy demands in
resent shortage of energy is expected to
'om possible end-use restrictions, energy
management are essentially required.
of energy in the building sector extends
ting, heating, cooling, and ventilation for
rt. The electricity consumption in the
estic sectors currently accounts for
nd is expected to grow at more than

ike in the cost of different forms of
growing importance of energy conserva-
tting costs in the building sector of the
yielding dividends, by way of reduced
conservation is also of vital national
ıports more than 50% of its petroleum
s and a major share of its foreign ex-
: to pay for these imports. Electricity,
t growth rate of all energy sources used,
ıpply in many parts. The shortage of peak
ıarticularly acute. Huge investments are
resh generation capacity. Conservation
ong way in alleviating the resources
supply sector ensuring a more produc-
:sources.
ɔn, which leads to more efficient use of
:ing comfort levels, does not mean ration-
· load shedding, but it is a means of
vasteful use of energy and taking action to
ıste.
g staff in commercial buildings know that
t part of their monthly expenditure and

some of them also make efforts to reduce or at least contain their energy costs. But how many of them can confidently claim that their building is operating at its peak level of energy efficiency? For that matter, how many engineering staff know what optimal level of energy efficiency their building is capable of achieving and how to achieve it and maintain that level? Best practice on energy management has changed significantly in the last decade. New technologies, new techniques, and new operating practices have shifted the target dramatically. Much higher levels of energy efficiency are now achievable and much greater savings and profits lie in store for commercial buildings.

After working on numerous projects related to energy conservation in offices, hotels, hospitals, shopping malls, flight kitchens, etc., TERI has acquired sufficient knowledge on various energy conservation methods. It is TERI's intention to disseminate this knowledge as widely as possible, especially to those who manage energy in commercial buildings. This publication of TERI, *Managing energy efficiency in hotels and commercial buildings*, is a humble effort in that direction. The book would enable readers to understand the energy consuming systems in commercial buildings and equipment used in these systems, and how to find out their operating efficiencies. This book also helps in finding out areas where energy wastage takes place and suggests appropriate energy-saving measures to optimize energy use in those areas. Case studies provided at the end of each chapter would help better understand what has been discussed in the chapter.

This publication is primarily targetted at professionals involved in energy management and energy auditing of commercial buildings. However, the contents of the book will be of interests and of benefit to practising system designers, energy consultants, students pursing degree in energy systems, and policy makers responsible for developing and enforcing good energy practices in their organizations. TERI also expects that this book, apart from being a handy reference document, will also inspire people to help optimize energy use in commercial buildings.

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Chapter 1

Electrical systems

Introduction

The major areas in the electrical systems in commercial buildings which have bearing on the energy cost, are (1) electric motors, (2) power factor management, and (3) diesel generating sets. A detailed understanding of these areas is essential for efficient energy management.

Electric motors

Polyphase (3-phase) AC (alternating current) induction motors account for most of the electricity used in the hotel industry. Their popularity stems from their relatively low capital investment, maintenance costs, and rugged design. This section focuses on polyphase, low-tension, squirrel-cage induction motors.

Motor selection and application

Electric motors are used to provide motive power to equipment such as pumps, blowers, and a variety of diverse equipment, which find application in the hotel industry. It is important that the user defines his need accurately to enable proper selection of a motor for a particular application. When selecting a motor for any application, the following points should be considered.

- System requirements in general, automatic/non-automatic controls, variable speed, etc.
- Technical aspects: breakdown torque, start-up torque, accelerating time, duty/load cycle, operating conditions
- Electrical system requirement
- Availability, reliability, inventory, and maintenance requirements
- Price of the motor.

It is frequently observed that motors having ratings higher than required for the given applications are used. There are several reasons for this. Original equipment manufacturers tend to keep a large safety factor in selecting the motor, which results in underloading the motor. Underloading the motor may also occur as a result of under-utilizing the equipment. Before going into some of the aspects of motor performance and means of reducing motor electricity consumption, a brief review of the factors affecting motor performance is given below.

Factors affecting motor performance

The efficiency of an electric motor is the ratio of the mechanical energy delivered at the rotating shaft to the electrical energy input at its terminals. The efficiency with which this transformation takes place is determined by intrinsic losses occurring within the motor. These losses can be reduced only by changes in the motor design.

Figure 1 shows the typical variation of efficiency, power factor, and input current with load for an induction motor. Part-load performance characteristics depend on the motor design and vary from motor to motor. For operating loads in the range of 60%–100% of rated load, the efficiency reduction is not very significant but the power factor drops considerably. On further reducing the load, both power factor and efficiency decrease, and the effect is significant at very low loads.

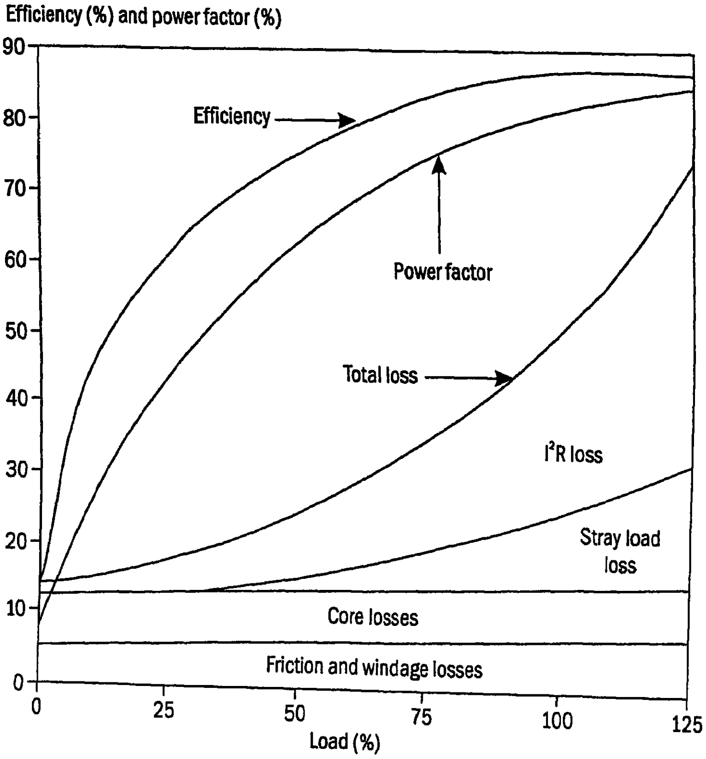

Figure 1 Typical loss distribution and approximate efficiency and power factor versus load for a 37-kW 3-phase induction motor

Intrinsic motor losses can be categorized into two groups—no-load or fixed losses and load or variable losses. No-load losses, as the term implies, occur even when the motor is unloaded, and are relatively constant over the entire load range. The fixed losses in a motor comprise magnetic core losses and friction and windage losses. Load losses vary with the motor load and consist of stator I^2R losses, rotor I^2R losses, and stray load losses. Magnetic core losses or iron losses consist of eddy current and hysteresis losses in the stator and rotor magnetic structure. They are independent of the load but depend on voltage, core steel material, and its physical geometry. Friction and windage losses are caused by friction in the bearings of the motor and windage loss of the ventilation fan and other rotating parts. These losses are also constant over the entire load range.

Copper losses (I^2R losses) are caused by currents flowing through a resistive path and are a function of the square of the current. Stray load losses are residual losses arising from a number of elements and are difficult to measure directly. They are proportional to the square of the rotor current. The factors that affect different types of losses in an induction motor are given in Table 1.

Table 1 Factors affecting different types of losses in a motor

		Copper loss				
Iron (hysteresis) loss	*Eddy current*	*Stator loss*	*Rotor loss*	*Friction loss*	*Windage loss*	*Load loss*
Material hot-rolled/ cold-rolled/cold-rolled non-grain-oriented steel	Conductivity of core steel	Operating load	Operating load	Bearing size and type	Type of enclosure	Operating load
Operating flux density	Operating flux density	Conductivity of winding material	Conductivity of winding/bar material	Weight of rotor	Heating produced due to other losses	Overhang of winding
Slot dimensions and wedges	Lamination thickness and uniformity	Cross-sectional area of winding	Cross-sectional area of winding/ bar	Type of lubricant	Type of fan: unidirectional/ bidirectional	Tooth harmonics
Stator-rotor slot combination (slot harmonics)	Core packing factor	Winding arrangement	Rotor length	Temperature of lubricant	Speed of rotor	Air gap flux density
Core packing factor	Inter-lamination insulation	Length of winding	Skin effect in winding	Alignment of shaft	Physical shape and dimensions of fan	Burning of laminations
Power frequency	Stator-rotor slot combination insulation	Skin effect in winding	Temperature of windings	Position of shaft - horizontal/vertical	–	Ratio of air gap to rotor slot opening
Slip at operating load	Slip at operating load	Temperature of windings	–	Speed of motor	–	Bonding of rotor bars to rotor laminations
Age of steel	Power frequency Age of steel	Slip of rotor	–	–	–	–

Table 2 Effect of voltage unbalance on motor performance

	Percentage unbalance		
Effect	0.3	2.3	5.4
Unbalance in current (%)	0.4	17.7	40
Increased temperature rise (°C)	0	30.0	40

Motor performance is also affected considerably by service conditions such as voltage and frequency, and voltage unbalance across the three phases. Voltage unbalance can be more detrimental to motor performance and motor life than voltage variation. Table 2 shows the effect of the voltage unbalance on the performance of the 5 h.p. motor. Another factor affecting efficiency is the practice of rewinding burned-out motors. Through proper rewinding, motor efficiency can sometimes be maintained at previous levels. However, in a majority of instances, poor workmanship and poor rewinding practices result in loss of efficiency.

Energy conservation in motors

Electricity consumption in motors can be reduced by operational improvements and retrofit improvements, which are listed below.

Operational improvements

- Operating at rated voltage and balanced supply
- Improving controls
- Regular maintenance
- Improving cooling.

Retrofit improvements

- Replacing oversized motors by motors of lower rating
- Replacing old and inefficient motors by new motors
- Replacing existing motors by energy-efficient motors
- Use of proper controls (e.g. variable speed drives and soft starters).

Perhaps the most significant savings in motor electricity could be effected by proper selection of the type and rating of motors. Underloading of motors has several disadvantages, namely (1) lower motor efficiency, (2) poor power factor, and (3) higher first cost of motor and control gear

For applications that have varying load patterns, it is not advisable to select a motor based only on the highest load to be taken by it. Instead, an optimum rating of motor may be selected, based on the loading cycle and the time for which the motor has to operate at each load during the cycle. The optimum rating in such cases is the equivalent steady-state rating, which would result in a temperature rise equal to that of the defined load cycle. The optimum rating may be calculated using the relation:

$$hp = \sqrt{\sum_{i=1}^{n} \frac{(hp_i^2 \times t_i)}{T_e}}$$

where t_i is the cycle time for which the motor operates at a load of hp_i, and T_e is the effective cooling time for the entire load cycle which is equal to t_i for the running period of the motor, and equal to one-fourth of t_i for the time the motor is not operating during the cycle (since the cooling is not effective during the motor off-time). It might be possible in many instances to select a motor of a slightly lower rating and operate at overload for a very short period of time (without exceeding the thermal capacity of the motor insulation), instead of selecting a motor of high rating to operate at full load for a short duration. This method of calculating the motor rating, however, is not suitable for motors subjected to frequent starts/stops and for motors operating high inertia loads.

Under operating conditions, however, it is normally quite difficult to ascertain the load on the motor to decide whether or not the motor is underloaded. The load on the motor can be determined only by measuring input parameters such as power, current, voltage, speed, and frequency, under both load and no-load conditions. It is not advisable to determine the load on the motor based on the motor load current alone, since this could lead to misleading results.

Field test of motors

Various international standards describe a variety of testing procedures to determine the efficiency of motors. In general, heavy testing equipment suggested in the standards are not suitable for conducting field tests in industrial units. The load test method and the equivalent circuit method are more practical in the industrial field environment. The load test requires a no-load test and load tests to be conducted at six load points ranging from one-fourth and one-and-a-half times the rated load. The equivalent circuit method requires a no-load test and a locked rotor test to be performed. The locked rotor test, however, requires a variable voltage power supply for conducting tests at lower voltages, which is not possible at a field site. For this reason, a modified form of the load test method and equivalent circuit method is used to determine the efficiency of the motor.

Estimation of efficiency of induction motors consists of three tests, namely resistance, load, and no-load tests. The resistance test is conducted twice, once at the start of taking field measurements and again after the load test. The load and no-load tests may be conducted in any sequence according to the convenience. In the case of standby motors, it is easier to conduct the no-load test before the load test. In the case of motors running under load, the load test is usually done first. However, due care has to be taken to ensure that the motor has cooled down before conducting the no-load test, so as to stabilize the stator resistance to its value at ambient temperature.

The results of the field measurement are analysed using a software, which estimates efficiency and power factor of the motor. The software also plots the efficiency and power factor curve of the motor at various loads. This helps in evaluating the performance of the motor and thus recommending suitable measures for improving the same. Figure 2 shows the average full-load efficiency of motors of different sizes as specified by the motor manufacturers and as observed in the field tests (corrected for full-load conditions). It is evident from the figure that motors are not operating at optimum loads or operating conditions.

Based on the results of such tests, it could be determined whether the motor could be replaced by a new motor (of the same rating), which would operate at a higher efficiency, or by

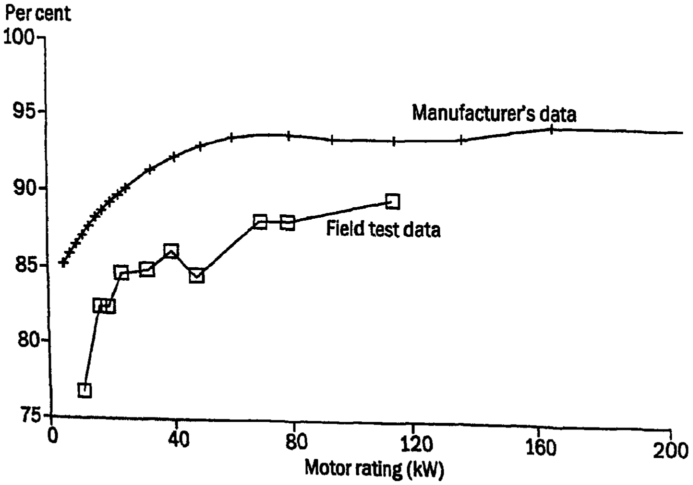

Figure 2 Full-load efficiency of motors of different ratings

a motor of lower rating, which would operate at an improved load factor, and consequently at an improved efficiency and power factor. An improvement in efficiency by even a few percentage points could result in significant savings depending upon the operating hours and the tariff structure. Lower input results not only in lower energy charges but also in lower demand charges due to reduced kVA (kilo volt amperes) demand.

Energy-efficient motors

Energy-efficient motors having higher efficiencies are obtained by incorporating design improvements to reduce intrinsic motor losses. Motor losses can be reduced by taking several measures, such as using low-loss steel, increasing the active material by incorporating a longer core length, using thinner laminations, reducing the air gap between stator and rotor, using copper bars in the rotor instead of aluminium and using superior bearings and a smaller fan. As a result of these modifications, costs of energy-efficient motors are also higher than those of standard motors.

The advantages of an energy-efficient motor are
- reduced power consumption,
- lower temperature rise and increased service life,
- broader band of constant efficiency thereby operating at light loads without appreciable drop in efficiency, and
- improved power factor of operation.

Energy-efficient motors having efficiencies higher by three to four per cent over those for standard motors are now available in the country. The power factor is also higher and a further advantage is a fairly flat performance characteristic over a wide range (typically 50%–100%) of loads. These motors cost about 30% more than equivalent standard motors. Even so, the savings that could be achieved at such improved efficiencies justify the additional expenditure incurred on the motor. The economics, however, depend on the operating hours and the electricity tariff. Usually, motors operating for less than 2000 hours a year are not good cases for replacement by energy-efficient motors.

The selection of an energy-efficient motor should be based on several factors additional to those for a standard motor.
- Electric power-saving and life-cycle-cost comparison with those of standard motors
- Improved ability to perform under adverse conditions such as abnormal voltage (generally, the energy-efficient motors have superior performance characteristics under abnormal voltage conditions)

- Lower operating temperature
- Lower noise level
- Ability to accelerate higher-inertia loads than standard motors.

In spite of higher operating efficiencies at all load points (at all loads, the energy-efficient motor presents an opportunity for energy savings), in certain applications and duty cycles, they cannot be justified on the basis of energy saved: (1) intermittent duty or special torque applications, fire pumps, etc. and (2) types of loads - multispeed, frequent starts and stops, very high-inertia loads, low-speed motors (below 720 rpm).

Case study

The case study explores the feasibility of using an appropriate rating motor in place of an existing 15 kW motor and works out the associated savings. The savings are worked out for both the standard motor and the energy-efficient motor.

Existing motor details

Rating : 15 kW
Full load current : 28 Amp
Full load speed : 2900 rpm
Full load efficiency : 0.865%
Full load power factor : 0.88

Existing motor operation

Motor loading : 55.3%
Peak current : 19.1 Amp
Efficiency : 0.84%
Power factor : 0.75
Load at shaft : 8.3 kW

Proposed option

Standard motor

Rating : 11 kW
Full load current : 21 Amp
Maximum loading (for the same load) : 77.2%
Motor efficiency at given load : 0.86%
Power factor at given load : 0.88

Associated energy savings

Reduction in input power : 0.229 kW
Annual energy savings* : Rs 4825
Net investment in replacement : Rs 13 100
Simple payback period : 2.7 years

Energy-efficient motor
Rating : 11 kW
Full load current : 20 Amps
Maximum loading for same load : 77.2%
Motor efficiency at given load : 0.9%
Power factor at given load : 0.88

Associated energy savings
Reduction in input power : 0.658 kW
Annual energy saving* : Rs 13 800
Net investment in replacement : Rs 24 100
Simple payback period : 1.75 years

*The savings are based on 8400 operating hours in a year and unit electricity cost of Rs 2.5/kWh.

Options for electricity saving in motors

Variable speed drives

Most motoring applications of electric machinery require either a constant speed or operation over a continuous or discrete range of speeds. A discrete number of speeds can be achieved by pole-change windings, which are obtained by switching the windings of the machine in banks/groups to produce different numbers of poles.

Continuously variable speed drives, on the other hand, have traditionally been the domain of DC machines. However, the emergence of solid-state frequency changers (inverters, cycloconverters, and the like) have permitted the operation of AC motors with adjustable speed.

Variable speed drives are increasingly finding applications in pumps and blowers. Variable speed drives offer considerable opportunities for energy savings for motor operation at low speed. The largest potential for electricity savings with variable speed drives is generally in variable torque applications, primarily in centrifugal pumps and fans, where the power requirement changes as the cube of speed. Apart from energy savings, the variable speed drives offer other advantages like soft starting facility, which reduces the mechanical stress in associated parts of the system.

Use of flat belts for transmission drives

Transmission drives, such as pulleys, belts, and gears, transmit mechanical rotational power from the prime mover shaft to a shaft connected to the mechanical load, such as a compressor, blower or a fan. The simplest form of a transmission drive is the belt drive, which is the least expensive as well as light and quiet in operation. The belt drive transmits power solely by the frictional grip between the belt(s) and the pulley. The belts consist of tension strands, which serve to absorb the tractional forces operating in the belts that are covered with an elastomer. The elastomer holds the strands together and provides surface friction for power transmission.

In many conventional belt drive systems, V-belts are normally used. New materials have made possible the manufacture of flat belts with better performance characteristics. Unlike V-belts that develop additional lateral frictional forces, a flat belt develops only the tangential forces. This reduces stress on the (flat) belt and results in lower heat generation due to lower lateral compression and elongation. Moreover, flat belts are free from the effect of heating at the edges that results in the hardening of rubber and reduction in the frictional grip. Due to the higher transmission efficiency of flat belts, savings in the range of 5%–12% can be achieved in the motor input power (10%–12% savings have been practically achieved in applications where flat belts are used). The payback period is normally less than a year.

Power factor

Power factor is defined as the ratio of real power (kW) to apparent power (kVA) and is the cosine of the angle by which the current lags (or leads) the voltage. In the case of inductive loads, the current lags the voltage and the power factor is lagging; whereas in the case of capacitative loads, the current leads the voltage and the power factor is leading.

Most industrial electric loads like motors and light sources using ballast are inductive in nature and the overall power factor of the plant is low and lagging. The poor power factor results in increased reactive current, which, in turn, increases the total current drawn by the system. The total current is the vector sum of the reactive and working components of the current:

$$I_t = \sqrt{(I_w)^2 + (I_m)^2}$$

where
I_t = total current,
I_w = working component of the total current, and
I_m = magnetizing component of the total current.

With an increase in the total current drawn, the I^2R losses in the line increase, which also results in greater voltage drop across the line leading to poor voltage regulation. Poor power factor also results in increased reactive power, leading to increased kVA demand by

$$kVA = \sqrt{(kW)^2 + (kVAR)^2}$$

where
kVA = apparent power,
kW = real component, and
kVAR = reactive component of power.

An improvement in power factor leads to the following advantages.

- Reduction of load on transformers and distribution equipment
- Reduction in load losses in transformers, distribution cables and other equipment, resulting in direct saving of kWh power consumption
- Stabilization of the system voltage
- Reduction in maximum demand, resulting in direct monetary savings
- Avoiding penalty imposed by the electricity board due to low power factor.

Methods of improving power factor

Poor power factor can be improved either by using shunt capacitors or by using synchronous motors that can be operated at leading power factor to compensate for loads with lagging power factor. Synchronous motors are very expensive and are used in very few industries. Therefore, the use of shunt capacitors is most suitable and widely acceptable for power factor improvement.

Installation of capacitors results in leading current and, thus, compensates for the lagging current drawn by the load. To determine the rating of a capacitor required for a particular load, it is necessary to determine the reactive power of the load. The capacitor rating required to improve the power factor from an existing value to a desired value can be computed using the equation

$$CkVAR = kW \times (\tan \theta_1 - \tan \theta_2)$$

where
CkVAR = capacitor kVAR required,
$\theta_1 = \cos^{-1}$ (existing power factor), and
$\theta_2 = \cos^{-1}$ (desired power factor).

Alternatively, power factor correction tables may be used to determine the rating of capacitors required to improve the power factor from an existing value to a desired value. Table 3 gives the kW multipliers for determining the capacitor kVARs

Table 3 Real component multipliers to determine the capacitor reactive component of power (kVARs) required for power factor improvement

	Cos θ_2[b]																				
Cos θ_1[a]	*0.80*	*0.81*	*0.82*	*0.83*	*0.84*	*0.85*	*0.86*	*0.87*	*0.88*	*0.89*	*0.90*	*0.91*	*0.92*	*0.93*	*0.94*	*0.95*	*0.96*	*0.97*	*0.98*	*0.99*	*1.0*
0.50	0.982	1.008	1.034	1.060	1.086	1.112	1.139	1.165	1.192	1.220	1.248	1.276	1.306	1.337	1.369	1.403	1.440	1.481	1.529	1.589	1.732
0.51	0.937	0.962	0.989	1.015	1.041	1.067	1.094	1.120	1.147	1.175	1.203	1.231	1.261	1.292	1.324	1.358	1.395	1.436	1.484	1.544	1.687
0.52	0.893	0.919	0.945	0.971	0.997	1.023	1.050	1.076	1.103	1.131	1.159	1.187	1.217	1.248	1.280	1.314	1.351	1.392	1.440	1.500	1.643
0.53	0.850	0.876	0.902	0.928	0.954	0.980	1.007	1.033	1.060	1.088	1.116	1.144	1.174	1.205	1.237	1.271	1.308	1.349	1.397	1.457	1.600
0.54	0.809	0.835	0.861	0.887	0.913	0.939	0.966	0.992	1.019	1.047	1.075	1.103	1.133	1.164	1.196	1.230	1.267	1.308	1.356	1.416	1.559
0.55	0.769	0.795	0.821	0.847	0.873	0.899	0.926	0.952	0.979	1.007	1.035	1.063	1.093	1.124	1.156	1.190	1.227	1.268	1.316	1.376	1.519
0.56	0.730	0.756	0.782	0.808	0.834	0.860	0.887	0.913	0.940	0.968	0.996	1.024	1.054	1.085	1.117	1.151	1.188	1.229	1.277	1.337	1.480
0.57	0.692	0.718	0.744	0.770	0.796	0.822	0.849	0.875	0.902	0.930	0.958	0.986	1.016	1.047	1.079	1.113	1.150	1.191	1.239	1.299	1.442
0.58	0.655	0.681	0.707	0.733	0.759	0.785	0.812	0.838	0.865	0.893	0.921	0.949	0.979	1.010	1.042	1.076	1.113	1.154	1.202	1.262	1.405
0.59	0.619	0.645	0.671	0.697	0.723	0.749	0.776	0.802	0.829	0.857	0.885	0.913	0.943	0.974	1.006	1.040	1.077	1.118	1.166	1.226	1.369
0.60	0.583	0.609	0.635	0.661	0.687	0.713	0.740	0.766	0.793	0.821	0.849	0.877	0.907	0.938	0.970	1.004	1.041	1.082	1.130	1.190	1.333
0.61	0.549	0.575	0.601	0.627	0.653	0.679	0.706	0.732	0.759	0.787	0.815	0.843	0.873	0.904	0.936	0.970	1.007	1.048	1.096	1.156	1.299
0.62	0.516	0.542	0.568	0.594	0.620	0.646	0.673	0.699	0.726	0.754	0.782	0.810	0.840	0.871	0.903	0.937	0.974	1.015	1.063	1.123	1.266
0.63	0.483	0.509	0.535	0.561	0.587	0.613	0.640	0.666	0.693	0.721	0.749	0.777	0.807	0.838	0.870	0.904	0.941	0.982	1.030	1.090	1.233
0.64	0.451	0.474	0.503	0.529	0.555	0.581	0.608	0.634	0.661	0.689	0.717	0.745	0.775	0.806	0.838	0.872	0.909	0.950	0.998	1.058	1.201
0.65	0.419	0.445	0.471	0.497	0.523	0.549	0.576	0.602	0.629	0.657	0.685	0.713	0.743	0.774	0.806	0.840	0.877	0.918	0.966	1.026	1.169
0.66	0.388	0.414	0.440	0.466	0.492	0.518	0.545	0.571	0.598	0.626	0.654	0.682	0.712	0.743	0.775	0.809	0.846	0.887	0.935	0.995	1.138
0.67	0.358	0.384	0.410	0.436	0.462	0.488	0.515	0.541	0.568	0.596	0.624	0.652	0.682	0.713	0.745	0.779	0.816	0.857	0.905	0.965	1.108
0.68	0.328	0.354	0.380	0.406	0.432	0.458	0.485	0.511	0.538	0.566	0.594	0.622	0.652	0.683	0.715	0.749	0.786	0.827	0.875	0.935	1.078
0.69	0.299	0.325	0.351	0.377	0.403	0.429	0.456	0.482	0.509	0.537	0.565	0.593	0.623	0.654	0.686	0.720	0.757	0.798	0.846	0.906	1.049
0.70	0.270	0.296	0.322	0.348	0.374	0.400	0.427	0.453	0.480	0.508	0.536	0.564	0.594	0.625	0.657	0.691	0.728	0.769	0.817	0.877	1.020
0.71	0.242	0.268	0.294	0.320	0.346	0.372	0.399	0.425	0.452	0.480	0.508	0.536	0.566	0.597	0.629	0.663	0.700	0.741	0.789	0.849	0.992
0.72	0.214	0.240	0.266	0.292	0.318	0.344	0.371	0.397	0.424	0.452	0.480	0.508	0.538	0.569	0.601	0.635	0.672	0.713	0.761	0.821	0.964
0.73	0.186	0.212	0.238	0.264	0.290	0.316	0.343	0.369	0.396	0.424	0.452	0.480	0.510	0.541	0.573	0.607	0.644	0.685	0.733	0.793	0.936
0.74	0.159	0.185	0.211	0.237	0.263	0.289	0.316	0.342	0.369	0.397	0.425	0.453	0.483	0.514	0.546	0.580	0.617	0.658	0.706	0.766	0.909
0.75	0.132	0.158	0.184	0.210	0.236	0.262	0.289	0.315	0.342	0.370	0.398	0.426	0.456	0.487	0.519	0.553	0.590	0.631	0.679	0.739	0.882
0.76	0.105	0.131	0.157	0.183	0.209	0.235	0.262	0.288	0.315	0.343	0.371	0.399	0.429	0.460	0.492	0.526	0.563	0.604	0.652	0.712	0.855
0.77	0.079	0.105	0.131	0.157	0.183	0.209	0.236	0.262	0.289	0.317	0.345	0.373	0.403	0.434	0.466	0.500	0.537	0.578	0.626	0.686	0.829
0.78	0.052	0.078	0.104	0.130	0.156	0.182	0.209	0.235	0.262	0.290	0.318	0.346	0.376	0.407	0.439	0.473	0.510	0.551	0.599	0.659	0.802
0.79	0.026	0.052	0.078	0.104	0.130	0.156	0.183	0.209	0.236	0.264	0.292	0.320	0.350	0.381	0.413	0.447	0.484	0.525	0.573	0.633	0.776

Continued

Table 3 *Continued*

	*Cos θ₂*ᵇ																				
Cos θ₁ᵃ	*0.80*	*0.81*	*0.82*	*0.83*	*0.84*	*0.85*	*0.86*	*0.87*	*0.88*	*0.89*	*0.90*	*0.91*	*0.92*	*0.93*	*0.94*	*0.95*	*0.96*	*0.97*	*0.98*	*0.99*	*1.0*
0.80	0.000	0.026	0.052	0.078	0.104	0.130	0.157	0.183	0.210	0.238	0.266	0.294	0.324	0.355	0.387	0.421	0.458	0.499	0.547	0.609	0.750
0.81		0.000	0.026	0.052	0.078	0.104	0.131	0.157	0.184	0.212	0.240	0.268	0.298	0.329	0.361	0.395	0.432	0.473	0.521	0.581	0.724
0.82			0.000	0.026	0.052	0.078	0.105	0.131	0.158	0.186	0.214	0.242	0.272	0.303	0.335	0.369	0.406	0.447	0.495	0.555	0.698
0.83				0.000	0.026	0.052	0.079	0.105	0.132	0.160	0.188	0.216	0.246	0.277	0.309	0.343	0.380	0.421	0.469	0.529	0.672
0.84					0.000	0.026	0.053	0.079	0.106	0.134	0.162	0.190	0.220	0.251	0.283	0.317	0.354	0.395	0.443	0.503	0.646
0.85						0.000	0.027	0.053	0.080	0.108	0.136	0.164	0.194	0.225	0.257	0.291	0.328	0.369	0.417	0.477	0.620
0.86							0.000	0.026	0.053	0.081	0.109	0.137	0.167	0.198	0.230	0.264	0.301	0.342	0.390	0.450	0.593
0.87								0.000	0.027	0.055	0.083	0.111	0.141	0.172	0.204	0.238	0.275	0.316	0.364	0.424	0.567
0.88									0.000	0.028	0.056	0.084	0.114	0.145	0.177	0.211	0.248	0.289	0.337	0.397	0.540
0 89										0.000	0.028	0.056	0.086	0.117	0.149	0.183	0.220	0.261	0.309	0.369	0.512
0.90											0.000	0.028	0.058	0.089	0.121	0.155	0.192	0.233	0.281	0.341	0.484
0.91												0.000	0.030	0.061	0.093	0.127	0.164	0.205	0.253	0.313	0.456
0.92													0.000	0.031	0.063	0.097	0.134	0.175	0.223	0.283	0.426
0.93														0.000	0.032	0.066	0.103	0.144	0.192	0.252	0.395
0.94															0.000	0.034	0.071	0.112	0.160	0.220	0.363
0.95																0.000	0.037	0.079	0.126	0.186	0.329
0.96																	0.000	0.041	0.089	0.149	0.292
0 97																		0.000	0.048	0.108	0.251
0.98																			0.000	0.060	0.203
0 99																				0.000	0.143
																					0.000

ᵃoriginal power factor; ᵇdesired improved power factor

Note For example: find the CkVAR to improve the power factor of a 500-kW load from 0.7 to 0.95; CKVAR = kW × multiplier = 500 × 0.691 = 345.5

required for power factor improvement. The recommended power factor range is between 0.94 and 0.96.

Case study A case study highlighting the saving on demand charges by installing capacitors and the investment required for the purpose is given in Table 4.

Table 4 Savings on demand charges by and investment required for installing capacitors

Parameter	*Case I*	*Case II*
Maximum demand of the unit (kW)	1000	1000
Power factor of the load without capacitor	0.8	0.8
Rating of the capacitor installed in Case II (kVAR)	–	500
Cost of the capacitors @ Rs 1000/kVAR	–	500 000
Power factor after installation of capacitor	–	0.97
Maximum demand of the unit (kVA)	1250	1031
Monthly demand charges @ Rs 170/kVA	212 500	175 270
Saving per year (Rs)	–	446 760
Payback period (months)	–	11

At a few places, industries have to pay a penalty to the power supply utility if the average power factor on a monthly basis is less than the specified level. An additional monetary saving could be achieved by avoiding this penalty if the power factor is improved by the concerned unit, resulting in a higher return on investment as a result of the capacitor installation.

Basis for capacitor selection

There are different types of capacitors available with respect to design (namely, metalized polypropylene, all polypropylene, mixed dielectric, twin layer, etc.) The selection of type of capacitor bank depends on the following points.

- Load condition and harmonic content of the power system
- Initial cost
- Life time
- Capacitor power loss
- Reduction in capacitor rating during operation (ageing)
- Voltage condition

In case the voltage variation and harmonics are not properly considered in the design of the capacitor unit, then their effect is to over-stress the dielectric material. Excessive harmonic currents flowing into the unit may cause overheating of the capacitor and subsequent failure.

Methods of power factor correction

There are three basic methods of capacitor power factor correction. For large installations, at least two or, if possible, all three methods are followed.

Manual control

A group of capacitors permanently connected to MCC (motor control centre). Wherever there is a load variation, the capacitors are manually switched on and off.

Auto control

Wherever there is a load variation, the capacitors required for improving the power factor will also change. Auto power factor control is the best method available to maintain the power factor within the set limit irrespective of the load variation.

Individual correction

Power factor of individual motor is improved by connecting individual capacitor banks in parallel. For higher range of motors, which are running continuously without much variation in load, individual power factor correction is advised.

Location of capacitor installation

Capacitors may also be provided at any one or all of the given three places to improve the power factor at different stages and avoid the high cost of installing low-rating capacitors:

1. together at main sub-station;
2. among distribution centres of power feeders and divided according to capacitor requirement; and
3. at the terminals of individual motor and/or inductive load.

Ideally, capacitors should be connected as near to the load as possible to get the maximum benefit, since improvements due to capacitors are reflected backwards from the point of installation to the generators. Therefore, by providing capacitors only at the sub-station, the consumer would avoid the penalty for poor power factor but would lose out on other benefits such as reduced line losses, reduced loading of cables, improved voltage regulation, and increased system efficiency. However, commercial buildings, which do not pay a tariff based on the kVA demand, may not find it economical to install capacitors at individual load points.

While installing capacitors, care should be taken while selecting proper capacitor rating. Over-compensation is equally harmful and may also give rise to excessive voltage build-up when the motor is switched off. To provide capacitor compensation at the motor's terminals, it should be ensured that the capacitor current is not greater than the no-load current of the motor or the capacitor kVAR is not greater than about 90% of the no-load kVAR of the motor. Similarly, when connecting capacitors at distribution panels, it must be ensured that loads are not switched off with all capacitors connected in circuit. Since there are load fluctuations in the plant, it would be extremely difficult to switch the capacitors on and off manually in order to keep the overall power factor high at all loads. To avoid the over-compensation, the power factor correction controller (relay) can be connected to the main panel that will automatically control the switching of capacitors from the capacitor bank, delivering high power factor (at a preselected level) at all loads.

In addition to the savings that are evident, there are other savings that are more difficult to quantify. For example, improved voltage regulation results in better operation of other equipment leading to an overall increase in system efficiency. In the economic analysis, however, the installation of capacitors is influenced by the tariff structure, the cost of capacitors, and the monetary savings that can be achieved.

Diesel generating sets

With the gap between the demand and supply of electric power widening, the role of DG (diesel generating) sets in the Indian hotels and commercial buildings cannot be overemphasized. Generally, DG sets are employed as a stand-by to meet a part or the full requirement of the unit in case of power failures.

In this section, the various aspects related to energy conservation in a DG set are discussed, with specific focus on assessing the operating performance and measures that can be undertaken to reduce fuel and oil consumption.

System description

In order to evaluate the performance of a DG set (specifically with regard to energy conservation), some understanding of the system is necessary. In this section, some relevant details about the system are provided to help the energy auditor appreciate the underlying concepts.

A DG set consists of a prime mover and an alternator. The prime mover is, invariably, a compression ignition engine, which, depending on its design, can run either on HSD (high speed diesel), LDO (light diesel oil), LSHS (low-sulphur heavy stock) or other heavier petroleum fractions. With the laying out of natural gas pipelines (or a gas grid as it is sometimes referred to), and gas being made available to industries, many installations use gas also, either in dedicated systems or along with diesel in dual-fuel operation. In the prime mover, the heat energy of the fuel is converted into mechanical energy through the rotary motion of the output shaft of the engine. The output shaft is directly coupled to an alternator, which generates the electric power. Therefore, in principle, the DG set consists of two main sub-systems: an IC (internal combustion) engine (most frequently, a compression ignition one) and an alternator. For all practical purposes, while conducting an energy audit of a DG set, it is the engine or the prime mover part, which is concentrated upon.

IC engines are generally considered to be highly energy efficient. This basically stems from the fact that the efficiency levels of these engines are substantially higher compared to their predecessor, the external combustion engines. But the fact remains that in spite of their relatively superior performance, the efficiency level of a typical engine is only about 26%. As can be seen from Figure 3, of the total energy input (in terms of fuel burnt), 34% is lost through the exhaust, 30% to the cooling medium (coolant), and only the remaining 36% is converted to useful work, defined as indicated power. Of this, another 10% is lost due to engine friction and others. Thus, out of 100 units of energy that goes into the system, only 26% is available at the output shaft. Although the figures given above may vary from one engine design to another, the fact remains that the efficiency level of these engines is quite

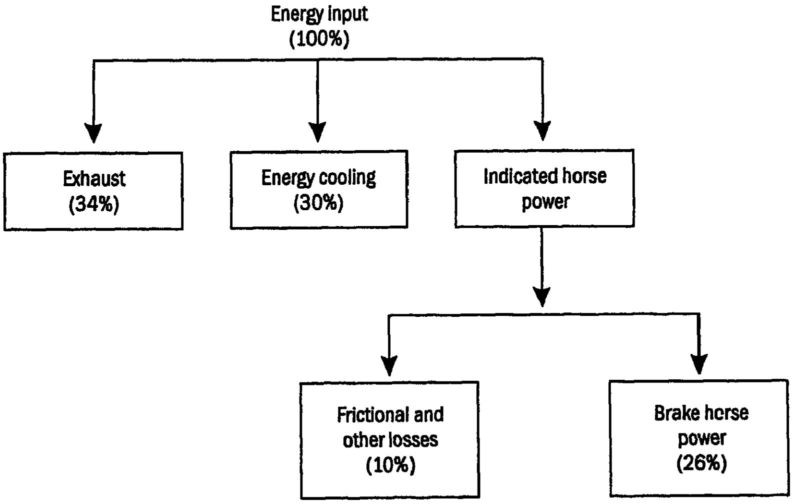

Figure 3 Typical energy distribution in an internal combustion engine

low as against the general belief. Some of the losses discussed above are, however, unavoidable and are quite peculiar to the basic engine design. There is also a limit to which the efficiency of an IC engine can be improved. The aim here is not to go into an in-depth analysis of the design, but to concentrate on items that can be controlled at the user's end to effect conservation of fuel and oil. In a system like the DG set, more often than not, the end user needs to contact the manufacturer for the remedy of a particular problem. On the users' part, identification of the problem, therefore, assumes a much greater role. In the subsequent sections, the performance assessment along with some conservation options for the fuel and lubricating oil usage in a DG set are discussed.

Fuel conservation

In a DG set, reduction of fuel consumption, even by a small percentage, can lead to enormous monetary savings. But the first step to approach the problem is to assess the operating efficiency of the DG set. This gives an idea as to whether there is a scope to effect savings, and if so, by how much. The efficiency of a DG set is the combined efficiency of the engine and the alternator. Since it is relatively difficult to compute the efficiency level of the sub-systems, the combined efficiency is generally computed. The overall efficiency is, therefore, defined as the number of units of electricity generated per litre of fuel consumption. The resultant figure is called SEGR (specific electricity generation ratio) with the unit of kWh/litre. Once the above is computed, it remains to be compared with the normative figure supplied by the DG set manufacturer. The recommended level of SEGR values for different sizes of DG sets are shown in Table 5.

The single most important component in the above exercise is the proper monitoring of fuel consumption for the DG set. It is absolutely essential to have separate fuel measurement arrangements for each individual DG set to arrive at the correct SEGR value. If the actual SEGR figure is found to be lower than the normative value recommended by the manufacturer, then effort should be made to identify the causes for this deviation. In case the operating efficiency of the DG set is lower than the recommended value, in spite of the engine loading and other operational parameters being within the limits and the periodic maintenance schedule also being followed, it is always advisable to take the help of the DG set manufacturer. The nature of the problem could range from improper fuel injection (due to clogged/damaged injectors, fault in the fuel-injection pump, etc.) to excessive wear and tear in the ring/liner assembly leading to high blow-by and

Table 5 Recommended specific electricity generation ratio values for different diesel generating set ratings

Rating (kVA)	*Recommended SEGR* (kWh/litre)
1450	3.84
1100	3.88
625	3.73
608	3.50
550	3.50
500	3.84
400	3.69
310	3.30
250	3.20
180	3.07
175	3.00
166	3.00
160	3.00
120	3.00

subsequent power loss. It is difficult to rectify some of these faults at the user-end and the help of the manufacturer becomes a necessity. There are areas, however, where the DG set user can himself take remedial actions. Some of these measures are discussed below.

Proper engine loading

The extent to which a DG set is loaded pronouncedly affects its operating efficiency. At part-load operations, the efficiency of the DG set drops with a consequent decrease in the specific energy generation ratio. The drop in efficiency becomes fairly pronounced at loads below 60% and is a function of (1) the ratio of actual output to rated capacity and (2) design and capacity rating of the DG set. Figure 4 illustrates a typical relationship between the extent of loading and the efficiency level, showing the inefficient range and the preferred range of operation.

Keeping the above in mind, one must always take enough care to ensure that, to the extent possible, the sets are optimally loaded. Specifically in cases where there are more than one or a bank of DG sets, proper load distribution among the sets can yield rich benefits.

Lubricating oil

As can be seen from Figure 3, frictional losses are low compared to the input of fuel energy, but they form a much more significant proportion of the total useful work. Concentrating

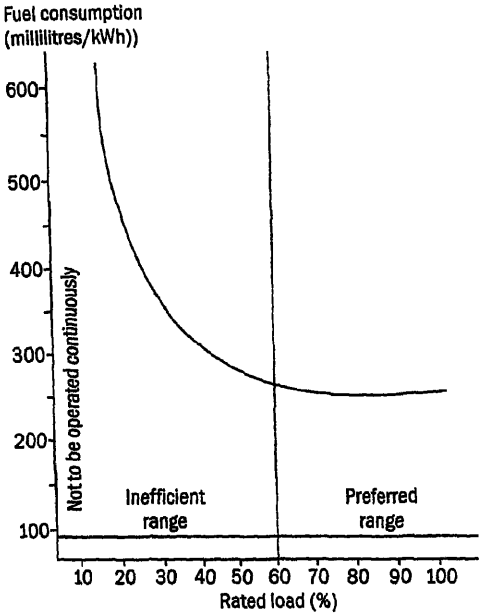

Figure 4 Effect of loading on the efficiency of a diesel generating set

our efforts on the mechanical losses, it is clear that lubrication is the prime area to improve operating efficiency. The choice of lubricant is, therefore, important and can help in reducing the frictional losses as against fuel consumption. The single most important parameter in this regard is the oil viscosity, which profoundly influences the frictional characteristics of a moving part. Although the viscosity grade of the oil to be used is specified by the DG set manufacturer, appropriate choices can be exercised in this regard. It is always preferable to use a multigrade oil in place of a monograde oil. Although the cost of multigrade oil is marginally higher than its monograde counterpart, the additional investment in shifting to multigrade oil is justified in terms of improved fuel efficiency. Without going into the details of the chemistry of oil formulation, the reason for better fuel efficiency with multigrade oil is briefly discussed below.

Multigrade oils, when first introduced, were primarily formulated to have a wider acceptable viscosity range compared to the monograde variety. This means that the multigrade oil was formulated in such a way that it neither becomes too viscous (too thick) at low temperatures nor very thin (low viscosity) at high temperatures. The benefits were obvious—these oils were able to operate over a wide range of temperature and avoided the use of (thereby maintaining an inventory) different grades of oil for different seasons. However, in addition to this primary duty, these oils were also found to provide fuel economy benefits due to the additives used in the formulation process. In simpler terms, these additives are polymeric long-chain chemical compounds. In the bearing areas, where there is a high rate of shear, these oils undergo temporary viscosity loss due to the sheer breakdown of the long-chain additives. This temporary viscosity loss leads to lowering of the viscous drag, eventually leading to an improvement in fuel economy. The extent of fuel saving possible by this route is about 1.5%.

Lubricating oil conservation

For engine applications, the lubricating oil should have special characteristics for desirable performance. The finished lubricants are, therefore, high-value products and need to be used efficiently. The three main ways to achieve lubricant conservation are (1) reduction in oil consumption, (2) re-refining of used oil, and (3) optimizing the drain interval.

Reduction in oil consumption is an engine-design-related subject and there is little that the DG set users can do at their end, except to keep a check on the mechanical condition of the engine. The major routes through which oil is lost are the valve stem to guide clearance and the ring packs. Heavily worn-out engines will obviously have high clearances, thereby increasing the oil loss. As a rule of thumb, oil consumption should not exceed one per cent of the fuel consumption, and for modern engines this value is still lower. If the oil consumption exceeds this limit, the engine manufacturer should be consulted. Re-refining of used oil is done by high temperature distillation and chemical treatment of oil or any other process which removes all undesirable physical and chemical contaminants so as to produce a pure base stock. But it can only be practised in cases where there are a large number of DG sets and the oil quantity handled is big enough to justify the setting up of a re-refining plant. The optimization of the drain interval is something that can be attempted at the user-end, selected details of which are presented below.

Optimizing oil drain interval

Crankcase oil needs renewal because it gets contaminated by the atmosphere, soot, unburnt fuel, and sulphurous acid from the blow-by products and also gets oxidized in service due to exposure to heat and air. The oil also loses its potential due to depletion of the functional additives. An optimum oil drain period represents the best compromise between the safety of engine parts and the economy, in terms of quantum of oil required over a specified period of operation. Oil in service is drained after fixed hours of operation, based on the recommendation of the engine manufacturer. These recommendations are broad-based, bearing very little correlation with the actual state of the used oil. So the situation calls for regular monitoring of the quality of oil in service and draining it only when it is found unfit for further use. The quality of the oil can be judged on the basis of the following parameters.

- Viscosity change
- Flash point
- Insoluble build-up
- Neutralization value
- Water content
- Silica content.

It is always advisable to consult the oil supplier to decide the discard limits of the above oil parameters for specific applications. The discard limits shown in Table 6 can be taken as a rough guide for 'HD Type 3 oil' (Servo Super grade of the Indian Oil Corporation Ltd).

Table 6 Rough guide for the discard limit of Type 3 oil

Oil characteristics	*Limit for change of engine oil*
Flash point	180 °C minimum
Viscosity change	±25% at 100 °C
Pentane insolubles	5% maximum
Water content	0.2% maximum
Total base number	1.5 minimum
Silicon	20 parts per million maximum

Some of the critical parameters of used oil could be monitored at regular intervals and the oil be discarded only when it reaches/approaches the discard limit. There are many instances both in stationary as well as in mobile applications, where the oil change based on the condition monitoring of used oil has yielded many benefits.

Fuel additives

Fuel additives are generally used to tackle specific problems and are resorted to when conventional methods fail. Typical examples of these are the use of carburettor cleanliness additives, in case of gasoline engines, and the use of additives to counter injector nozzle choking in diesel engines. The claims with regard to additives that can improve energy efficiency in an engine should be weighed carefully. There are fuel efficiency improvement additives available in the market, with the manufacturers' claims of reducing the fuel consumption by as much as 10%–15%. One must review these claims with extreme caution since practical experience has shown that the additives are invariably not cost-effective. The Petroleum Conservation Research Association carries out assessment/ evaluation of such additives routinely, and it is always preferable to take their advice in this regard. Alternatively, the additive supplier should be asked to submit the test/evaluation report being carried out by an accredited laboratory.

Waste heat recovery

As already explained, about 35% of the total energy input in a DG set is lost through the exhaust gases. Most of the industrial sets of higher rating are, however, turbocharged versions, where part of this energy is utilized in running a turbine, which, in turn, runs a compressor, delivering supercharged air (above atmospheric pressure) to the engine. The resultant effect is an improvement in the specific output of the engine. Another major advantage of supercharging is in the case of installations at higher altitudes. In these cases, the drop in engine output due to thinner air at a higher altitude is compensated by ramming in more air under this mode. In spite of this, the heat loss through the exhaust gases is quite substantial and can be utilized in specific cases. The recovered heat wasted could be used for one or more of the applications listed below.

- Pre-heat combustion air
- Hot water generation
- Waste-heat boiler for steam generation
- Vapour absorption refrigeration system

One thing, however, must always be kept in mind. The waste heat recovery from the DG set is possible only when large quantities of it are available on a continuous basis. This implies that for DG sets that are run only as a stand-by, waste-heat utilization options will not be economically viable. Only in cases where the set is run on a more or less continuous basis, one can expect a fairly reasonable payback period.

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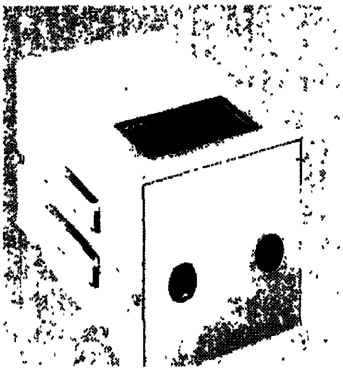

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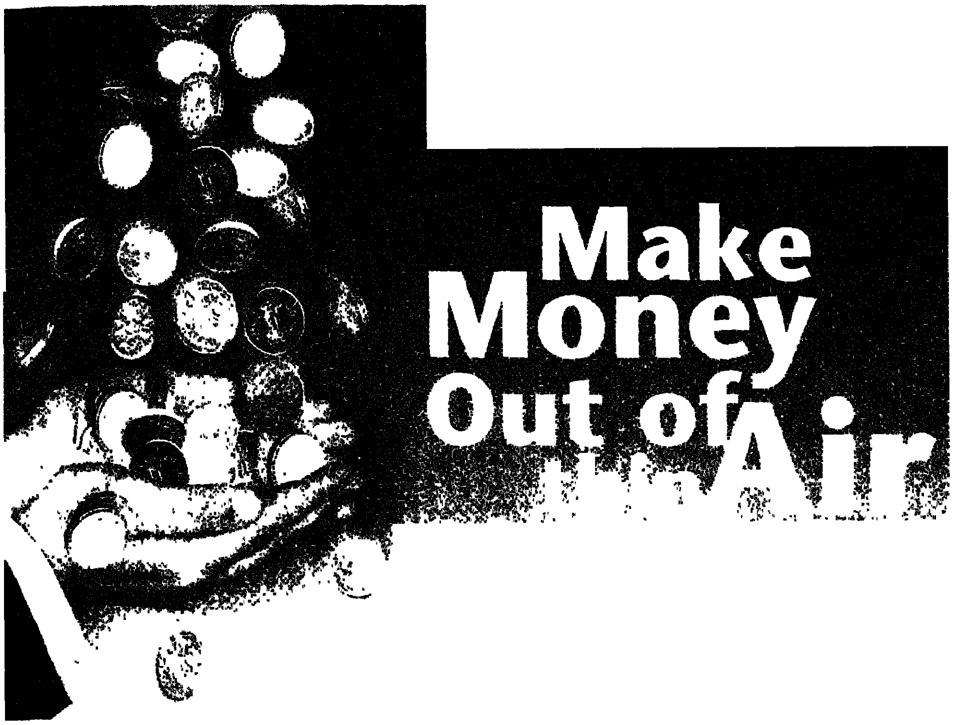

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Chapter 2 Steam and hot water systems

Introduction

Steam boilers are used in various hotels and commercial buildings to convey heat for different applications. Boilers are used to generate small, medium, and large quantities of hot water in hotels and commercial buildings. Steam is commonly used as the heating medium mainly for two reasons: (1) it is generated from soft water, which is usually available through a softener; and (2) it is able to store a large quantity of heat (at a high temperature) which can be conveniently used. Hot water is regularly consumed at different places in hotels and commercial buildings. There are many alternative methods used to generate hot water, depending on applications. The hot water generated from either normal (raw) water or soft water in the boiler is either directly consumed or circulated through the HWG (hot water generator), heat exchanger/calorifier installed; and the hot water in the secondary side sent to end-use systems in the hotel. The boilers with calorifier are considered to be on the primary side of the hot water system. The heat exchanger/calorifier in the secondary circuit is used to generate large quantity of hot water for two reasons: (1) if the hardness of raw water is very high; (2) if adequate storage of hot water is required for the non-working period of the hot water boiler. A fixed quantity of soft water is polished in a separate polishing unit in the hotel everyday. The polished water is heated in the hot water boiler (at the primary circuit) and continuously circulated through the heat exchanger/calorifier. Steam, independently or jointly with hot water, is used to generate hot water in many hotels. The coil arrangement of water heating in the calorifier/tank with steam and hot water is also provided separately in many hotels. Normally, the coils are placed inside the calorifier and the HWG for heating the water. Various types of fuels, such as coal, oil, gas, and biomass are used for the generation of steam and hot water in boilers, depending on the availability of fuel and the cost economics prevailing in the hotel and commercial building. Electrical heating systems are preferred for heating small quantities of water.

Steam boilers can be categorized into different types depending on water/flue gas passage in the boiler, fuel usage, and pressure generation. The types of steam boilers vary with respect to the requirement of the hotels and commercial buildings. The types of hot water generating system in hotels vary with respect to the quality of raw water, quantity of consumption, and requirement of heat/storage. The calorifier, in combination with the hot water boiler, is not commonly used in commercial buildings. Whatever may be the type of the boiler and HWG used, the motive of the hotels and commercial buildings should be to generate the required quantity and quality of steam and hot water with the maximum possible efficiency. This can only be achieved by reducing the various avoidable heat losses occurring within the boiler and HWG system, thus improving the efficiency of the same.

Different boilers will have different efficiency levels, depending on the fuel type. Table 1 shows the efficiency levels of different fuel-fired boiler existing in hotels and commercial buildings. From the table, it can be concluded that the one using gas as fuel will be the most efficient.

Table 1 Efficiency levels of different boilers

Boiler type	*Efficiency* (%)
Gas-fired	88-92
Oil-fired	80-86
Coal-fired	60-82

The types of steam boiler, hot water boiler, and other hot water generating system installations would vary with respect to the requirement of the unit. Once the raw water quality, hot water temperature, and heating capacity are known, a suitable selection can be made from the water heating system ranges available with manufacturers. Whatever be the type of boiler used, the motive of the unit should be to generate the required quantity and quality of hot water at minimum possible costs.

For units where raw water quality is poor and steam or high/medium pressure hot water is distributed throughout the year in large quantities for process requirements, the provision of steam/high temperature hot water heated storage calorifier is invariably most economical. For units where raw water quality is good and the high/medium/low pressure hot water of adequate quantity is required at desired pressure, the provision of compact/package type HWG is invariably the most economical.

Efficiency evaluation of boilers

There are two methods of evaluating the efficiency of boilers: (1) direct method and (2) indirect method. These are discussed in detail in the following sections.

Direct method

This is also known as the input–output method because it requires only the useful output and the heat input (fuel) to evaluate the efficiency. The formula required is:

$$\text{BE (boiler efficiency)} = \frac{\text{Heat output}}{\text{Heat input}}$$

For steam boilers, BE =

$$\frac{\text{Steam flow rate} \times (\text{Steam enthalpy} - \text{Feedwater enthalpy})}{\text{Fuel firing rate} \times \text{Gross calorific value}}$$

For hot water boilers, BE =

$$\frac{\text{Hot water flow rate} \times (\text{Hot water enthalpy} - \text{Feedwater enthalpy})}{\text{Fuel firing rate} \times \text{Gross calorific value}}$$

The direct method offers the following advantages.

- It is excellent for plant people to evaluate the efficiency of boilers quickly.
- It requires only a few parameters for computation.
- It needs minimal instrumentation for monitoring.

However, this method suffers from not being able to calculate losses under different heads to indicate why the efficiency is low.

Indirect method

The disadvantages inherent in the direct method can be overcome by the indirect method, which calculates various heat losses associated with the boiler. The efficiency can be arrived at by subtracting the heat loss percentages from 100. This method is also known as the heat loss method. The different heat losses occurring in a boiler and HWG are as follows.

- Dry flue gas loss, and heat losses due to moisture in fuel, due to moisture from burning hydrogen, due to moisture in the air, due to the presence of combustibles in refuse, due to carbon monoxide formation, due to sensible heat in bottom ash, and due to structure (surface radiation and convection) are common to steam and hot water boilers; heat loss due to blowdown is additional for steam boilers.

- Heat losses through structure are for HWG, heat exchanger/calorifier/hot water tanks only.

Dry flue gas loss (L_{fg})

Normally, the major energy loss in a boiler occurs through flue gases, which escape at a high temperature. Dry flue gas loss depends on two factors: (1) flue gas temperature and (2) quantity of flue gas generated (depending on the excess air level). The excess air is related to the fuel being used in the boiler. Typically, the following levels of excess air (Table 2) can be practised for different fuels used in the hotel to minimize the flue gas loss. Table 3 shows the theoretical air requirement and the theoretical CO_2 produced in the case of complete combustion.

Table 2 Typical excess air levels

Fuel type	*Excess air level* (%)
Coal	40-60
Oil	10-15
Gas	5-10

Table 3 Theoretical air and CO_2 level of fuels

Fuel	*Theoretical air requirement* (kg per kg air)	*Theoretical CO_2 level* (%)
Coal	6.3	18.70
Fuel oil	14.0	15.80
Natural gas	14.8	11.55

$$A_t = 11.51\ C + 34.30\ (H - O/7.937) + 4.335\ S$$

where,

A_t = theoretical air in kilogram required to completely burn a kilogram of fuel

C = carbon content obtained from laboratory analysis (kg/kg of fuel)

H = hydrogen content obtained from laboratory analysis (kg/kg of fuel)

O = oxygen content obtained from laboratory analysis (kg/kg of fuel)

S = sulphur content obtained from laboratory analysis (kg/kg of fuel)

EA = $(O_2 - CO/2) / (0.2682 \times N_2 - (O_2 - CO/2)$

where,

EA = excess air (fraction)

O_2 = percentage of oxygen in flue gas

CO = percentage of carbon monoxide in flue gas

CO_2= carbon dioxide in flue gas (%). If CO_2 is not measured by the instrument, it can be calculated using the following formula:

CO_2= $(1 - O_2/21) \times$ maximum theoretical CO_2

Maximum theoretical CO_2 in 1 Oil = 15.5%
2 Coal = 18.5%
3 Gas = 11.7%

N_2 = percentage of nitrogen in flue gas. This can be computed as the sum-total of the O_2, CO, and CO_2 percentage subtracted from 100. CO_2 is a derived quantity to be computed as shown in the section below.

$M_a = (1 + EA) \times A_t \times M_f$

where,

M_a = air flow rate (kg/h)

M_f = fuel consumption (kg/h)

$M_{fg} = M_a + M_f$

where,

M_{fg} = Quantity of dry flue gas (kg/h)

$L_{fg} = M_{fg} \times C_p \times (T_g - T_i)$

where,

L_{fg} = heat loss due to heat in dry flue gas (kcal/h)

C_p = mean specific heat of dry flue gas over the range between gas temperature leaving the unit and the ambient temperature (~ 0.25 kcal/ kg°C)

T_g = flue gas temperature (°C)

T_i = inlet air temperature (°C). To be considered as ambient temperature if no air preheater is being used.

Heat loss due to moisture in fuel (L_{mf})

$L_{mf} = M \times M_f \times (H_o - H_i)$

where,

M = moisture in fuel (kg/kg of fuel)

H_o = enthalpy of vapour at partial pressure, and exit gas temperature (T_g), to be determined from steam tables. For practical purposes, partial pressure approximates 1 atm. at steam generator exit (kcal/kg).
H_i = enthalpy of saturated liquid at ambient temperature. To be used only for solid and liquid fuels (kcal/kg).

Heat loss due to moisture from burning hydrogen (L_h)

$$L_h = (8.937 \times H \times M_f) \times (H_o - H_i)$$

where,
H = hydrogen in fuel (kg/kg of fuel)
8.937 kg of water is produced from burning 1 kg of hydrogen.

Loss due to moisture and hydrogen in fuel is dependent on the final flue gas temperature besides fuel composition (hydrogen and moisture content). The moisture content of fuel oils and gas is generally insignificant. The final flue gas temperature is therefore the only variable factor for a given fuel (other than for moisture in solid fuel).

Heat loss due to moisture in air (L_{ma})
Heat is absorbed by water vapour accompanying the air into the combustion chamber. Water vapour present in the air depends on the dry and wet bulb temperatures prevailing at the time.

$$L_{ma} = (M_{ma} \times M_a \times M_f) \times (H_o - H_{im})$$

where,
M_{ma} = kg of water vapour per kg of dry air at the measured DBT (dry bulb temperature) and the WBT (wet bulb temperature)
H_{im} = enthalpy of saturated vapour at DBT and WBT

Heat loss due to combustibles in refuse (L_{uc})
This generally applies only to solid fuel-fired boilers where unburnt fuel could be present in bottom ash or fly ash. A small loss due to unburnt carbon would also occur while burning fuel oils, but this is generally ignored in any heat account.

$$L_{uc} = M_{dr} \times M_c\, M_f \times 7837.5$$
$$M_{dr} = A / (1 - M_c)$$

where,

M_{dr} = dry refuse (kg/kg of fuel)

A = ash content in fuel (kg/kg of fuel)

M_c = unburnt carbon in refuse (kg/kg of refuse). If flue dust and bottom ash refuse differ materially in unburnt carbon content, they should be estimated separately using an ash balance.

7837.5 = heat of formation of CO_2 from carbon (kcal/kg of carbon)

Heat loss due to formation of carbon monoxide (L_{co})

Carbon monoxide appears in the flue gas due to incomplete or partial combustion. As CO can further combine with O_2 to form CO_2, its presence represents an energy loss.

$$L_{co} = [CO / (CO + CO_2)] \times C_b \times M_f \times 5636.7$$
$$C_b = C - M_{dr} \times M_c$$

where,

CO = carbon monoxide in flue gas (%)

C_b = carbon burnt (kg/kg of fuel)

5636.7 = heat of formation of CO_2 from CO (kcal/kg of carbon)

Heat loss due to sensible heat in bottom ash (L_{sa})

$$L_{sa} = M_{ba} \times M_f \times C_{pa} \times (T_{ba} - T_a)$$

where,

M_{ba} = bottom ash quantity (kg / kg of fuel). To be determined by performing an ash balance between ash in coal, fly ash and bottom ash

C_{pa} = specific heat of bottom ash (~ 0.25 kcal/kg°C)

T_{ba} = bottom ash temperature (°C)

T_a = ambient Temperature (°C)

Blowdown loss (L_{bd})

In steam boilers, the effect of water containing some dissolved solids and removal of pure steam results in a concentration of solids in the water. Above a certain level of concentration, these solids encourage foaming, causing carry-over of water into the steam, and could cause scaling inside the boiler leading to tube failure because of overheating. This concentration has to be reduced and, therefore, some boiler water is removed and make-up water added. The blowdown is at a high temperature and represents a heat loss. However, with good water treatment and heat recovery, the overall heat loss could be well under two per cent of the heat input.

$$L_{bd} = M_{bd} \times (H_{bd} - H_{fw})$$

where,

M_{bd} = quantity of blowdown (kg/h)
H_{bd} = enthalpy of saturated liquid at drum pressure (kcal/kg)
H_{fw} = enthalpy of feedwater (kcal/kg)

Loss due to surface radiation and convection (L_{st})

Radiation and convection losses occur from the exterior region of the boiler. It can be around one per cent or less of the heat input at maximum rating. It may, however, be considerably higher in older boilers, and can be as high as 10% where the insulation is in poor condition and the boiler's design is old. The loss in actual heat units is generally constant, so that, as a proportion of the heat input, the percentage loss is lowest when on 'full fire' and maximum on 'low fire'.

Structural loss = S_r (radiation loss) + S_c (convection loss)

$$S_r = B \times E \times A \times (T_s^4 - T_a^4)$$

where,

B = Stefen Boltzmann constant
E = emissivity of the surface
A = surface area
T_s = surface temperature
T_a = ambient temperature

$$S_c = h \times A \times (T_s - T_a)$$

where,

h = heat transfer coefficient

Total heat loss = $L_{fg} + L_{mf} + L_h + L_{ma} + L_{uc} + L_{co} + L_{sa} + L_{bd} + L_{st}$

Thermal efficiency of boiler (%) =

$$\frac{(\text{Heat input per hour} - \text{total heat loss per hour}) \times 100\%}{\text{Heat input per hour} \times 100}$$

Case Study 1

Boiler details

Boiler design efficiency, % : 82.0
Steam flow rate, TPH : 51
Steam pressure, kg/cm^2 : 41
Steam temperature, °C : 315
Feed water flow rate, TPH : 53
Feed water temperature, °C : 100
Fuel : low sulphur heavy stock
Fuel consumption, kg/h : 4000

Ultimate analysis of LSHS (%)

Carbon : 86.70
Hydrogen : 11.87
Sulphur : 1.00
Oxygen : 0.00
Ash : 0.14
Moisture : 0.29
Gross Calorific Value, kcal/kg : 10 424

Flue gas analysis

Oxygen, % : 7.0
Carbon monoxide : 70 PPM (parts per million)
Temperature : 300 °C

Temperature (°C)

Dry bulb : 35
Wet bulb : 25

Structural details

Average emissivity : 0.2
Average surface temperature, °C : 70
Total surface area, m^2 : 700

Solution

Dry flue gas loss (L$_{fg}$)

Stoichiometric air requirement, kg of air/kg of fuel : 14.1
Excess air, % : 46
Flue gas quantity, kg/h : 86344
Specific heat of flue gas, kcal/kg °C : 0.245
Sensible heat in dry flue gas, million kcal/h : 5.60
Total heat input, million kcal/h : 41.69
Dry flue gas loss, % : 13.44

Heat loss due to moisture in fuel (L_{mf})

Enthalpy of vapour at boiler exit, kcal/kg : 734
Enthalpy of saturated liquid, kcal/kg : 25
Heat loss due to moisture in fuel, kcal/h : 8225
Heat loss, % : 0.020

Heat loss due to moisture from burning hydrogen (L_h)

Heat loss due to H_2 in fuel, million kcal/h : 3.00
Heat loss, % : 7.2

Heat loss due to moisture in air (L_{ma})

Absolute humidity, kg water vapour/kg dry air : 0.02
Enthalpy of saturated vapour, kcal/kg of water vapour : 18
Heat loss due to moisture in air, million kcal/h : 1.18
Heat loss, % : 2.83

Heat loss due to formation of carbon monoxide (L_{co})

Carbon burnt, kg carbon/h : 3468
Heat loss due to formation of CO, million kcal/h : 0.0132
Heat loss, % : 0.031

Blowdown loss (L_{bd})

Feed water TDS, PPM : 2
Blowdown TDS, PPM : 55
Blowdown rate, % : 3.77
Feedwater flow rate, kg/h : 53 000
Flow rate of blowdown water, kg/h : 2000
Enthalpy of blowdown water, kcal/kg : 261
Enthalpy of feedwater, kcal/kg : 100
Heat loss due to blowdown, million kcal/h : 0.322
Heat loss, % : 0.77

Loss due to surface radiation and convection (L_{st})

Emissivity : 0.2
h for horizontal plate facing upwards : $1.885 \times (T_s - T_a)^{0.25}$
h for horizontal plate facing downwards : $0.976 \times (T_s - T_a)^{0.25}$
h for horizontal vertical plate : $1.465 \times (T_s - T_a)^{0.25}$
Radiation losses, kcal/h : 50545
Convection losses, kcal/h : 86963
Total structural losses, million kcal/h : 0.1375
Losses, % : 0.32

$$\text{Efficiency} = 100 - (L_{fg} + L_{mf} + L_{h} + L_{ma} + L_{uc} + L_{co} + L_{sa} + L_{bd} + L_{st})$$
$$= 75.38\%$$

Direct method

Useful heat gained by steam, million kcal/h	: 31.44
Heat input, kcal/h	: 41.69
Efficiency, %	: 75.39

Energy conservation measures

After calculating the thermal efficiency of a boiler, it is easier to pinpoint the areas of energy loss in steam and hot water boilers and to suggest further steps to minimize them.

ECM (Energy Conservation Measure) 1

Maintaining optimum excess air level in steam and hot water boiler

The combustion air supply to the steam and hot water boiler is always kept more than the theoretical (stoichiometric) requirement for proper mixing of fuel and its complete combustion. If the air supply is less than the theoretical requirement of the fuel, it would lead to the generation of unburnt fuel. On the other hand, if the air supply is very much in excess, a large quantity of heat would be carried away in flue gas. Thus, an optimum level of excess air level should be maintained to minimizing the flue gas losses. Figure 1 shows the relationship between excess air, O_2, CO_2 and CO.

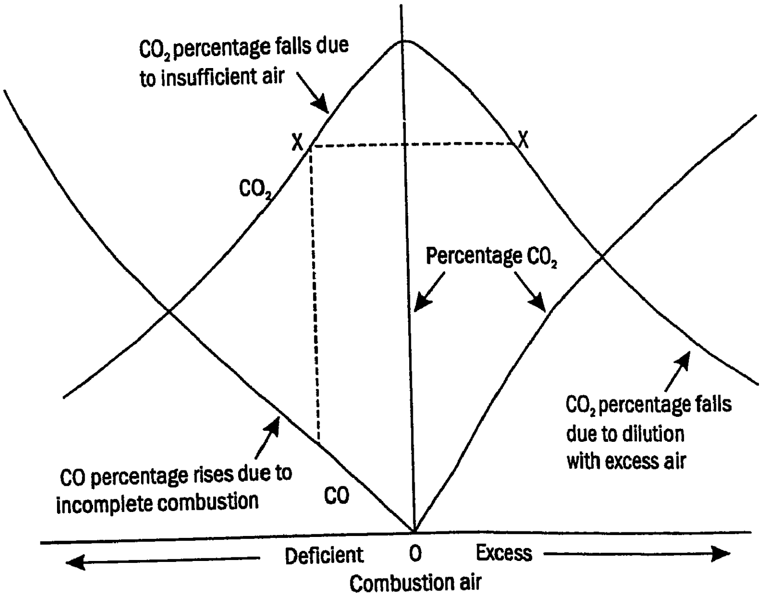

Figure 1 Excess air, oxygen, carbon dioxide, and carbon monoxide combustion

ECM 2 **Improvement in the return of condensate in a steam boiler, unused hot water in a hot water boiler/calorifier**

It is important to return the condensate to the steam boiler and unused hot water to the hot water boiler as quickly as possible in ordinary circumstances. Returning the condensate and unused hot water can also reduce fuel consumption. Recycling of the condensate will increase the net steam output of the boiler as heated feedwater will require less heat to achieve the outlet pressure and temperature condition. As a thumb rule, every 5 °C rise in feedwater temperature reduces fuel consumption throughout in the boiler by one per cent. The return condensate/hot water carries sensible heat, which can be as high as 20% of the total heat input to the steam and hot water boilers. Besides, it will also save the cost of make-up water, which would otherwise have been necessary to replace the discarded condensate/hot water.

ECM 3 **Optimization of blowdown in steam boiler**

Though carrying out blowdown is required, depending on the water quality and steam pressure, the normal tendency in a steam boiler is to do more blowdown than the requirement. This results in loss of sensible heat of boiler water (at boiler pressure). The rate of blowdown must be adjusted in such a way that the concentration of solids is kept close to the design level. This would help in minimizing the boiler losses due to blowdown.

ECM 4 **Waste heat recovery in steam boiler**

Usage of heat recovery system

If the steam boiler is not provided with a heat recovery system, either an economizer or an air preheater, the possibilities of installing one can be worked out. As a thumb rule, every 20 °C decrease in flue gas temperature would improve the efficiency of the steam boiler by about one per cent.

Performance of heat recovery system

It is also equally important to maintain the performance of the heat recovery systems. Each system is designed to give a performance level that suits the steam boiler conditions. The inlet and outlet conditions of the cold and hot fluids must be monitored, and any deviation from the design level should be analysed for bad performance. For example, any fouling on the heat transfer surfaces of the heat exchanger would adversely affect its performance.

ECM 5 Improvement in the insulation of steam boiler, hot water boiler and hot surface of tanks

The structural loss of the boiler normally accounts for one to three per cent of the heat input. Though this figure is small, it is possible to minimize this loss further by upgrading the insulation material that may have worn out over a period of time. The surface of steam boilers, hot water boilers and hot water tanks are normally insulated with loose glass wool. Generally, the cylindrical surfaces of a boiler are provided with glass wool insulation 2.5" – 3.5" thick, and aluminium cladding. The cover plates (either at top or front, rear end) of most boilers are bare.

Radiation and convection losses depend on the thickness of insulation, type of mattress used, and the cladding material used. The structural heat loss for a large boiler will be less as the surface area exposed per unit heat input is less. Loose mineral/rock wool made into a mattress results in a higher heat loss compared to lightly resin bonded mineral/rock wool mattress.

The heat loss should be reduced significantly using resin bonded, rock wool fibres of different thicknesses. The temperature limit of two insulating materials used in hotel and commercial building is given in Table 4.

Table 4 Detail of typical insulating materials

Insulation	*Type*	*Density* (kg/m^3)	*Temperature limit* (°C)
Glass fibre	Fibrous	10-150	550
Rock wool	Fibrous	20-250	800

While aluminium (type III and grade I with low emissivity) cladding may result in a higher outer casing temperature compared to a GI (galvanized iron) sheet cladding, but heat loss with aluminium cladding will be lower than that in the GI cladding for the same thickness and type of insulating material.

Hot water heating system as a new installation or possible conversion of an existing steam heating system

The smaller systems, ranging from 5–15 MM Btu/h are only marginally more economical to operate than steam, but they are favoured because they provide much more accurate and uniform temperature control. Large segments of the chemical process industries have been benefiting from such conversions, by having hot water systems supplied for process heating. The advantages of installing hot water systems or of converting existing steam-heating systems to hot water in hotels and commercial buildings should be reviewed on a case to case basis. The disadvantages of steam heating and the advantages of hot water heating are given below.

Disadvantages of steam heating

Apart from fuel savings, the following situations contribute to heat losses in steam-heating systems: open vents on condensate receivers; flash out losses and leaks in steam traps; and boiler blowdowns. These losses involve at least an additional 5% in fuel costs. Another disadvantage is boiler deterioration, caused by instantaneous changes from low fire to peak loads. This is practically eliminated by high temperature hot water systems, because their 'flywheel' effects dampen those of peak loads.

A hot water system, on the other hand, is a closed circulating loop, with only very minor losses from leakage at valve stems and at pump stuffing boxes. In as much as the amount of make-up water required is only a minor/reasonable fraction of that required for a steam system, this eliminates the need for blowdowns and steam trap management, thereby doing away with the source of considerable heat loss.

Condensate lines are usually subject to corrosion, because mineral-free water, combined with atmospheric oxygen, becomes very corrosive. Condensate collection systems usually flow by gravity, so all lines are pitched in the direction of the receiver. Forced-circulation hot water lines are entirely independent of the plant layout.

Advantages of hot water heating

For new installations, the total capital investment is about the same for both steam and hot water systems. However, the savings involved in fuel costs and maintenance make the payback period for a new hot water system shorter than for the conversion of an existing steam system.

Many plants use their steam boilers for both direct steam consumption and indirect heating in process. The greater heat capacity of hot water over steam at equivalent saturation temperatures and the narrower pipelines required are other advantages. A steam line must be many times larger than an equivalent hot water line, which need only be one or two sizes larger than the condensate line required in the steam system.

The make-up water volume decreases in a properly designed hot water and return system supplies in comparison to the make-up water to the team boilers and condensate system with minimum chance of condensate flashing, thereby, eliminating steam-condensate flash out losses.

As compared to heat capacity, steam yields up about 800-960 Btu/lb when it condenses, while HTW can transfer only 100-150 Btu/lb. This, however, is no basis for comparison. Consider, for example, a 1 ft-long, standard 6-inch diameter pipe filled with 150 psi (g) steam. This pipe will hold 0.073 lb of steam with a heat capacity of roughly 90 Btu. This same pipe with water at 366 °F (saturation temperature corresponding to 150 psi (g) will hold 11 lb/ft, which contains about 3800 Btu. Hence, the ratio of absolute heat-storing capacity is 42 to 1 in favour of water. A graphical comparison of heat contents for different pipe diameters is given in Figure 1.

An HTW distribution system acts as a heat accumulator due to its capacity to store heat. This may be likened to an energy reservoir, which can accommodate sudden heat demands without loss in temperature. Steam systems often suffer a temperature drop when shock or peak loads occur, which cause a drop in the boiler pressure. But HTW generators need not be sized for maximum loads because of the heat stored. Steam boilers, however, require such sizing to prevent pressure losses and accompanying temperature losses. Smaller pipe diameters for HTW systems are economically important when heat is transported over long distances. Smaller pipes also reduce insulation costs.

One argument preventing installations from switching to hot water says that steam lines are needed for certain production methods, and that such steam would be slightly difficult to obtain from hot water.

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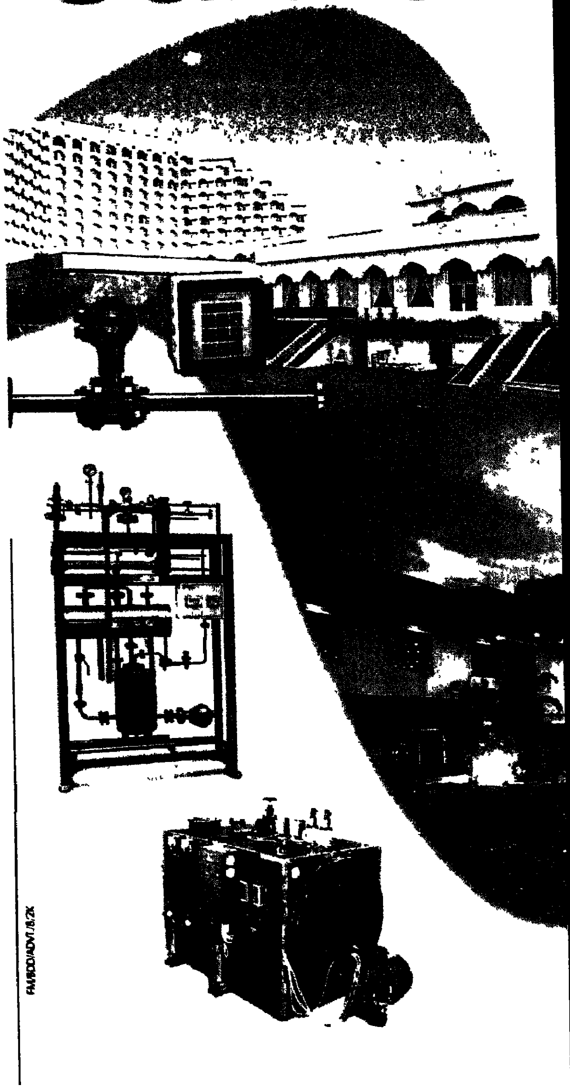

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THERMAL INSULATION OF BUILDINGS

Thermal Insulation System is a term used for entire combination of materials and not only for a single material (insulant). However, it is very important how insulation and ancillary materials are applied. Even the best insulating material fails in service if ancillary materials are not chosen properly and applied correctly.

Today there is hardly any building which is not air-conditioned and do not feature thermal nsulation as an engineered provision for the roof and ceiling. There is always a debate concerning the optimum positioning of the thermal insulation system for roofs and ceiling of buildings.

Choices available are :

a) Provision of insulation as a contact treatment to the inner surface of the roofing. This is normally termed as "Under-deck" insulation. Such treatment can be done either by using Rockloyd Resin Bonded Rockwool Slabs/ Rolls or Polyisocyanurate Slabs which are termed as Class-I fire rating materials when tested as per BS : 476 (Pt.7).

b) Provision of Insulation over a Suspended Ceiling (False Ceiling) system with Rockloyd Resin Bonded Rockwool slabs. Now-a-days there is trend to have Mineral Fibre Acoustical False Ceiling Tiles laid with metallic grid which has insulation value and also looks very nice.

c) Provision of insulation over the roofing structure – normally referred to as "Over-deck" insulation. Preferred insulation materials qualifying this application are :

1. High density expanded Polystyrene slabs
2. Extruded Polystyrene slabs
3. Rigid Polyurethane / Polyisocyanurate Foam sprayed in-situ directly over the roof.

The above application is followed by a waterproof protective treatment. Incase of Polyurethane foam an elastomeric water proofing treatment of minimum 1mm thick moisture – cure urethane (Isothane Waterproofing) based treatment is applied directly over the foam, forming a homogenous, jointless system spread across the roof. Incase of slabs first a layer of cement plaster has to be provided before the waterproofing treatment.

For metallic roof buildings now a days preinsulated sandwich panel – the one step insulated roofing and cladding is available. These panels comprise of a profiled sheet on top and a linear tray at the bottom with high density polyurethane foam sandwiched in between. These panels can be directly fixed to the purlins.

By K.K. MITRA
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Chapter 3 Refrigeration and air-conditioning system

Managing energy efficiently in hotels and commercial buildings 47-88

Refrigeration and air-conditioning systems have a large impact on the energy performance of commercial buildings. Not only do they consume 40% to 60% of the total commercial buildings' electricity, but they also have a large influence on the occupants' behaviour and building profitability. Energy conservation studies conducted in the past indicate that air-conditioning systems receive less attention than they deserve during planning, designing, operating and maintenance. Many cost-effective opportunities to improve the energy efficiency of systems are frequently overlooked. However, it remains a real challenge for building staff responsible for handling these systems to optimize energy consumption without compromising on the required thermal comfort within the building.

Air-conditioning system load in a typical luxury hotel

Studies indicate that the air-conditioning systems contribute 55% of the total connected load in a typical luxury hotel in India, which comes out to 42.5 W/m^2 of the total built-up area. In installed air-conditioning systems, compressors contribute to the maximum load (62%), chilled water pumps account for 10%, condenser water pump for 9%, cooling towers for 3%, air-handling units for 12%, and remaining 4% is fan coil units load (Figure 1).

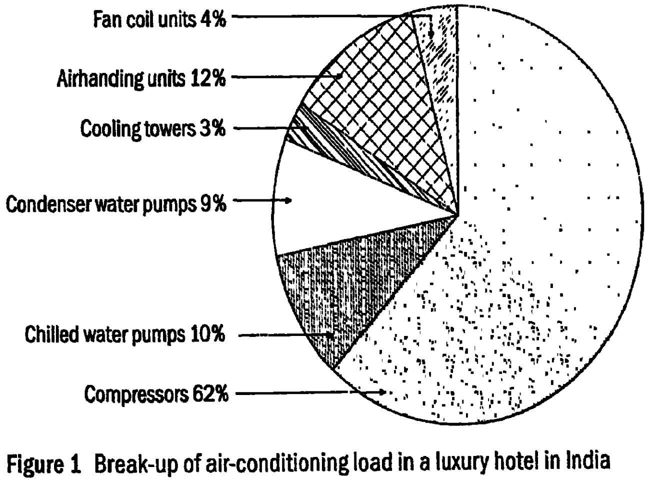

Figure 1 Break-up of air-conditioning load in a luxury hotel in India

Air-conditioning system energy consumption in a typical luxury hotel

Energy consumed in air-conditioning systems is around 45% of the total electricity consumed in a luxury hotel or 82 kWh/m^2 of the total built-up area. Within the air-conditioning system, compressors consume maximum energy (61%), chilled water pumps and condenser water pumps account for 11% and 7%, respectively, while the air-handling units and cooling towers consume 13% and 3%, respectively, of the total air-conditioning system input electrical energy. Remaining 5% energy is used in fan coil units (Figure 2).

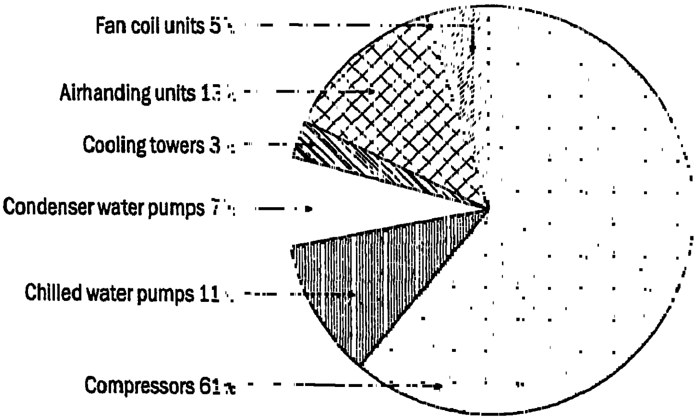

Figure 2 Break-up of air-conditioning energy consumption in a luxury hotel in India

Refrigeration systems

Vapour compression systems

Vapour compression refrigeration is the primary method used to provide mechanical cooling. All vapour compression systems consist of four basic components: evaporator, condenser, compressor, and an expansion device. The evaporator and condenser are heat exchangers that evaporate and condense the refrigerant while absorbing and rejecting heat. The compressor takes the refrigerant vapours from the evaporator and raises the pressure sufficiently for the vapour to condense in the condenser. The expansion device controls the flow of condensed refrigerant at this high pressure back into the evaporator.

Evaporator

The evaporator typically cools either air returning from the building space (or outside air) to 55–60 °F or water from about 54 °F as it returns from building air handlers to 44 °F. The refrigerant is circulated through numerous parallel paths. As the refrigerant flows and evaporates along these paths the pressure drops as well. This in turn drops the temperature of the refrigerant as it evaporates. Properly designed direct expansion coils operate with the coldest refrigerant temperatures closest to the coil exit. However, the refrigerant temperature coming out of this coil is usually a little warmer than this to provide some level of superheat to ensure that the liquid refrigerant is not leaving the coil and entering compressor, which may cause mechanical failure. Shell and tube exchangers commonly have water circulated through the tubes and refrigerant boiling around the tubes. There are also designs where refrigerant flows within the tubes and water flows over the tubes. Baffles are normally used in this case to direct water flow in a serpentine fashion to optimize heat transfer. Almost all-large chillers use shell and tube evaporators with water flowing through the tubes.

In comfort cooling applications, actual cooling loads are seldom at full load conditions. Capacity control is achieved in finned coil evaporators that directly chill air by splitting the coil into independent sections. The principal reason is to permit coil sections to be activated and deactivated to better match coil cooling capacity with compressor loading. The combination of smaller coil sections controlled by correspondingly sized expansion valves improves valve performance and part load humidity control.

Capacity control in shell and tube evaporators is usually handled using the return water temperature. For example, if the full load temperature range for chilled water is from 44 °F to 54 °F, water returning at 50 °F indicates that the cooling load is about 60%. Liquid refrigerant is metered to the evaporator to match the load using an orifice plate system or an expansion valve. On large chillers, the expansion valve is pilot operated.

Condenser

The refrigerant is recovered by condensing it in a condenser using air or water to reject the heat. Air-cooled condensers are most common in smaller sizes, up to about 200-ton capacity. Technically, there is no upper limit on the size of an air-cooled condenser, but operating cost issues dictate water-cooled units for applications over about 100 tons. There are two water-cooled designs: cooling towers and evaporative cooling process depend upon the wet bulb temperature of the air entering the unit, the volume of air flow, and the efficiency of the air/water interface. Evaporative condensers use water sprays and air flow to condense refrigerant vapours inside the tubes. The condensed refrigerant drains into a tank called a liquid receiver. Refrigerant sub-cooling can be accomplished by piping the liquid from the receiver back through the water sump where additional cooling reduces the liquid temperature even further. Cooling towers are essentially large evaporative coolers where the cooled water is circulated to a remote shell and tube refrigerant condenser. The cooling water is circulated through the tubes while refrigerant vapour condenses and gathers in the lower region of the condenser. This area also subcools the refrigerant below the temperature of condensation by bringing the coldest cooling tower water into this area of the condenser. The warmed cooling water is sprayed over a fill material in the tower. Some of it evaporates in the moving air stream. The evaporative process cools the remaining water. The volume of water used by both evaporative condensers and cooling towers is significant. Not only does water evaporate just to reject the heat, but water must be added to avoid the build-up of dissolved solids in the basins of the evaporative condensers or cooling towers. If these solids build up to the point that they foul the condenser surfaces, the performance of the unit can be greatly reduced.

Electric chillers

Reciprocating compressors

Reciprocating compressors are reciprocating piston-type compressors and generally manufactured in three types.

1 *Hermetic* Compressor-motor assembly contained in a welded steel case, typically used in household refrigerators, residential air-conditioners, smaller commercial air-conditioning and refrigeration units.

2 *Semi-hermetic* Compressor-motor assembly contained in a casting with no perpetration by a rotating shaft and with gasketed cover plates for access to key parts such as valves and connecting rods.

3 *Open* Compressor only with shaft seal and external shaft for coupling connection to belt-or-direct-drive using as electric motor or natural gas engine. These are largely used for ammonia refrigeration applications as hermetic designs cannot be used with ammonia refrigerant, and for engine-driven units.

In reciprocating units, as the piston nears the bottom of its stroke within the cylinder, the intake valve opens and the refrigerant vapour enters. As the piston rises, the increased pressure closes the intake valve. Then as the piston nears the top of its stroke, the exhaust valve opens permitting the vapour at the higher pressure to exit. Reciprocating compressor capacity is a function of the bore and stroke of the piston-cylinder configuration as well as the speed of the machine. Compressor capacity is also related to the compression ratio.

The mechanical design is rugged and reliable but has one significant limitation. Reciprocating compressors are designed to handle vapours, not liquids. When liquid enters the cylinder on the intake stroke, it tends to damage the valves on the compression stroke and possibly the compressor itself. This is why chillers incorporate liquid-to-suction on heat exchangers, which assure some level of vapour superheat at the compressor suction. Capacity is controlled by multiple staging of smaller compressors or in large multiple cylinder reciprocating compressors by unloading banks of cylinders on the compressors making machine most efficient at full load. Therefore, for maximum efficiency the reciprocating machines should generally be operated at full load.

Screw compressors

Helical rotary (or screw) compressors are positive displacement machines and available in two types: single-screw and twin-screw. A twin-screw compressor consists of accurately matched rotors (one male and one female) that mesh closely when rotating within a close tolerance common housing. One rotor is driven while the other turns in a counter-rotating motion.

A single-screw compressor uses a single main screw rotor meshing with two gate rotors with matching teeth. The main screw is driven by the prime mover, typically an electric motor. The screw-like grooves gather vapours from the intake port, trap them in the pockets between the grooves and compressor housing, and force them to the discharge port along the meshing point path. This action raises the trapped gas pressure to the discharge pressure. If the power input is adequate and pressure differential between outlet and inlet pressures is within the design range of the machine, the screw compressor delivers the appropriate refrigerant gas volume. The refrigerant gas enters and exits the compressor through ports; not valves like reciprocating compressors. The mating rotors rotate at such close tolerances that they require cooling and lubrication. This is provided by forcing oil into the compressor at strategic points. The oil also acts as a seal for rotor-to-rotor and rotor-to-housing clearances.

Screw compressors are most often driven by a constant speed electric motor and screw compressor being a positive displacement machine, moves a fixed volume of refrigerant making refrigeration capacity control difficult. The design uses a slide valve that opens to vent some gas back to the suction port, reducing both the net gas flow and power input.

Several manufacturers offer packaged water chillers using screw compressors. The size of water-cooled units ranges from

50 tons to over 1200 tons. They normally use HCFC-22 and HFC 134a as refrigerants in space cooling designs and ammonia in process refrigeration. They employ hermetic or semi-hermetic designs for higher efficiency. Air and evaporatively cooled models can be used from about 60 to 350 tons and can use open-drives. In smaller sizes, packaged water chillers with screw compressors compete with reciprocating chillers and in larger sizes they compete with centrifugal chillers.

Centrifugal compressors

Centrifugal compressors use one or more rotating impeller to increase the refrigerant vapour pressure from the chiller evaporator enough to make it condense in the condenser. Unlike the positive displacement, reciprocating, scroll or screw compressors, the centrifugal compressor uses the combination of rotational speed, and tip speed to produce this pressure difference. The refrigerant vapours from the chiller evaporator are commonly pre-rotated using variable inlet guide vanes. The consequent swirling action provides extended part-load capacity and improved efficiency. The vapours then enter the centrifugal compressor along the excess of rotation. The vapour passageways in the centrifugal compressor are bounded by vanes extending from the compressor hub, which may be shrouded for flow-path efficiency. The combination of rotational speed and wheel diameter to creates the necessary tip speed to accelerate the refrigerant vapour to the high pressure discharge where they move on to chiller condenser. Due to their very high vapour-flow capacity characteristics, centrifugal compressors dominate the 200 ton and large chiller market. Centrifugal chillers are most commonly driven by electric motors, but can also be driven by steam turbines and gas engines. Depending on the manufacturers' design, centrifugal compressors used in water chiller packages may be 1-, 2-, or 3-stages and use a semi-hermetic motor or an open motor with shaft seal.

Packaged water-cooled centrifugal chiller compressors are available in sizes ranging from 85 tons to over 5000 tons. They used HCFC-123, HCFC-22, and HFC-134a as their refrigerants. This usually calls for semi-hermetic design, with single or multi-stage impellers. Two manufacturers (Carrier and McQuay) offer semi-hermetic gear-driven models. Trane offers multi-stage direct-drive semi-hermetic units. York offers an integrated open-drive geared design.

Scroll compressors

Scroll compressors use one stationary and one orbiting scroll to compress refrigerant gas vapours from the evaporator to the condenser of the refrigerant path. The upper scroll is stationary and contains the refrigerant gas discharge port. The lower scroll is driven by an electric motor shaft assembly imparting an eccentric or orbiting motion to the driven scroll. That is, the rotation of the motor shaft causes the scroll to orbit, not rotate about the shaft centre. This orbiting motion gathers refrigerant vapours at the perimeter, pockets the refrigerant gas, and compresses it as the orbiting proceeds. The trapped pocket works progressively towards the centre of the stationary scroll and leaves through the discharge port. Scroll compressors are a relatively recent compressor development and will eventually replace reciprocating compressors, since they achieve higher efficiency, have better part-load performance, and operating characteristics.

Vapour absorption systems

The absorption cycle uses a heat-driven concentration difference to move refrigerant vapours (usually water) from the evaporator to the condenser. This high concentration side of the cycle absorbs refrigerant vapours. Heat is then used to drive off these refrigerant vapours thereby increasing the concentration again. Lithium bromide is the most common absorbent used in commercial cooling equipment, with water used as the refrigerant. Smaller absorption chillers sometimes used water as the absorbent and ammonia as the refrigerant. The absorption chiller operates at very low pressure for the water to vapourize at a cold enough temperature (e.g. at 40 °F) to produce 44 °F chilled water.

In evaporator water at 40 °F evaporates off the chilled water tubes, thereby brings the temperature down from the 54 °F being returned from the air handlers to the required 44 °F chilled water supply temperature. One ton of cooling evaporates about 12 pounds of water per hour in this step. This water is absorbed by the concentrated lithium bromide solution due to its hygroscopic characteristics. The heat of vaporization and the heat of solution are removed using cooling water at this step. The solution is then pumped to the concentrator at a higher pressure where heat is applied, to drive off the water and thereby re-concentrate the lithium bromide. The water driven off by the heat input is then condensed (using cooling tower water), collected, and then flashed to the required low temperature to complete the cycle.

Absorption chillers are available in two types: (1) single-effect (stage) units using low pressure (20 psig or less) as the driving force, (2) double-effect (2-stage) units are available as gas-fired (either direct gas firing, or exhaust gas from a gas engine or turbine) or steam-driven with high pressure steam (40–140 psig).

Two-stage absorption chillers

The energy efficiency of absorption can be improved by recovering some of the heat normally rejected to the cooling tower circuit. A 2-stage absorption chiller accomplishes this by taking vapour driven off by heating the first stage concentrator (or generator) to drive off more water in a second stage. Two separate shells are used. The smaller is the first stage concentrator. The second shell is essentially the single-stage absorption chiller, containing the concentrator, condenser, evaporator, and absorption chiller. The temperatures, pressures and solution concentrations within the larger shell are similar to the single-stage absorption chiller as well. Steam at pressure typically in the 125–150 psig range is brought into the stainless steel tubes of the first-stage concentrator causing the solution there to boil. The pressure at which boiling occurs and the pressure of the released refrigerant vapour is approximately 5 psig. The partially concentrated solution from this stage flows through the high temperature heat exchanger where it is cooled by the lower temperature dilute solution returning from the concentrator. This concentrate then passes into the lower pressure second-stage concentrator where the vapours from the first-stage take into the final desired concentration levels. The second-stage operates at a pressure of 0.1 atmosphere. The re-use of the vapours from the first-stage generator makes the machine approximately 30% more efficient than single-stage absorption chillers.

Direct-fired absorption chillers

Direct-fired absorption chillers utilize a burner as the heat input for the absorption cooling cycle. Most operate either on natural gas or fuel oil. Since the heat input is at a very high temperature, they achieve a very high efficiency for the absorption cycle (something approaching 12 000 Btu of fuel input for each ton-hour of cooling output). The absorption cycle itself is virtually identical to that of the two-stage steam absorption chillers. However, unlike most steam absorption chillers, the direct-fired absorption chiller lends itself fairly readily to chiller-heater applications where both cooling and heating are achieved in the same unit. Direct-fired absorption chillers require a stack to vent combustion products which is not necessary in a steam-fired unit. In addition, the first costs of direct-fired units are higher than steam-driven units.

Absorption chillers use more water as they must reject more heat and require larger cooling towers and are more difficult than electric chillers to put on-line (start up) and to take off-lying (shut down) as they require a dilution cycle.

Table 1 Recommended chillers type for different cooling range

Range (tons)	*Condenser type*	*Rank*	*Chiller type*
1-100	All	1	Scroll
	All	2	Screw
	All	3	Centrifugal
101-300	All	1	Scroll
	All	2	Screw
	All	3	Centrifugal
	Water	4	Absorption (double effect)
	Water	5	Absorption (single effect)
301-1250	All	1	Centrifugal
	All	2	Screw
	Water	3	Absorption (double effect)
	Water	4	Gas Engine
	Water	5	Absorption (single effect)
1251-2000	All	1	Centrifugal
	All	2	Centrifugal
2001 and above	All	1	Centrifugal

System performance measures

Performance of the electric systems

Since the vast majority of cooling equipment is driven with electric motors, the most common performance measure is kW/ton. This can be calculated for any given chiller by simply taking the manufacturers' estimated power use and dividing by the cooling system capacity expressed in tons. For example, if the manufacturer estimates the peak power required for a 250-ton chiller is 150 kW, the chiller will use 150 kW/250 tons or 0.6 kW/ton on the design day. But what is included in this 150 kW? Is it the chiller alone, or the pumps and fans in the remainder of the system? If this power estimate was for a packaged rooftop unit supplying air to the building, the power estimate is clearly the total kW/ton. However, in buildings where a centrally located water chiller is used, there are also chilled water and condenser water pumps as well as cooling tower fans. Were these included in the kW/ton performance measure? Not usually! And, in these cases, they must be added in to get the total kW/ton performance.

Design professionals usually evaluate chiller performance by comparing kW/ton for the chiller itself plus the heat rejection circuit. After all, for any given building, the chilled water distribution system alternatives would be identical. The differences between low and high efficiency cooling system alternatives show up in the kW/ton for the chiller plus the heat rejection circuit. Therefore, this total should be used when comparing the kW/ton for chiller alternatives such as air cooled versus water-cooled, low versus high efficiency, and gas versus electric driven.

Coefficient of performance for electric systems

The term coefficient of performance (CoP) is simply the ratio of the cooling effect produced expressed in Btu/hour divided by the energy input also expressed in Btu/hour. For an electric chiller at 0.6 kW/ton, this ratio is 12 000 Btu for a ton of cooling divided by the corresponding 0.6 kW energy input. The units of kW can be best thought of as kW hours per hour. Each kW is equivalent to 3413 Btu, therefore 0.6 kW is 2048 Btu. Therefore, a 0.6 kW/ton chiller is equivalent to a CoP of 12 000 Btu/2048 Btu, about 5.9 CoP. There is a short-cut formula to compute CoP directly from any given kW/ton. Simply divide 3.516 by the chiller's kW/ton to derive the CoP (3.516/0.6 = 5.9).

Energy-efficiency ratio and seasonal energy-efficiency ratio

The cooling equipment used in residential and commercial buildings often expresses cooling system efficiency in term of the EER (energy-efficiency ratio) and/or SEER (seasonal energy-efficiency ratio). These are defined by the cooling effect (in Btu) divided by the power use (in watts) for the peak day (EER), or the seasonal average day (SEER). For example, a chiller operating at 0.6 kW/ton would have an EER of 12 000 Btu divided by 600 watts or 20. This is mathematically equivalent to multiplying the CoP by 3.41 or vice versa. Therefore, a chiller operating at 0.6 kW/ton is equivalent to a COP of 5.9, or an EER of 20.

Reciprocating compressors

Water-cooled reciprocating compressors at ARI (Air-conditioning and Refrigeration Institute) rating conditions (44 °F leaving chilled water, 85 °F entering condenser water) operate at around 40 °F evaporating and 100 °F condensing temperatures equivalent to pressures. They have an EER of about 15, equal to 0.79 kW/ton. However, in air-cooled conditions, the condensing pressure is likely to run up to a 130 °F temperature corresponding to pressures, with EER, ranging from about 10.4 up to 11.3, which equate to 1.15 to 1.06 kW per ton.

Assembled into chiller packages in the 20 to 200-plus ton capacities, air-cooled units will typically have EERs ranging from 9.0 to 10.9, equal to 1.33 to 1.10 kW per ton with an average of about 1.22 kW per ton. Similarly, water-cooled chiller packages will have EERs ranging from 13.1 to as high as 15.8, which equates to 0.92 to 0.76 kW per ton with an average of about 0.82 kW per ton.

Screw compressors

At ARI standard rating, a modern water-cooled screw compressor has an EER in the range 14 to 17, equal to 0.85 to 0.70 kW per ton at full load. Since chillers do not operate at full load often, screw compressors are more efficient at part load. Integrated part load performance for screw units can be as low as 0.42 kW per ton.

Air-cooled chiller packages with screw compressors in the range of 20 to 200 ton perform at about 9 to 10 EER, equivalent to 1.15 to 1.06 kW/ton with an average of about 1.1 kW per ton.

Centrifugal compressors

At ARI standard rating conditions centrifugal chiller's performance at full design capacity ranges from 0.53 kW per ton or lower to 0.68 kW per ton. This performance includes the semi-hermetic refrigerant cooled or open-type compressor motors.

Performance of fuelled system

Absorption systems have energy performance measures similar to electric-driven equipment.

Coefficient of performance

The CoP is the ratio of the cooling effect produced expressed in Btu/hour divided by the energy input expressed on the same terms. Most high efficiency, two-stage absorbers achieve a CoP of ~1, that is, they use about 12 000 Btu of fuel energy to produce one ton of cooling. However, the absorber has much higher heat rejection power use than a comparable electric-driven machine. And these fuelled systems also use electric energy for pumps, fans, and controls. Manufacturers generally do not count the power used in their CoP data.

Single-effect units typically have a CoP of 0.7, however, double-effect units have a CoP of 1.0 to 1.2.

Integrated part load value

Large chillers whether they are electric or heat activated, usually perform the best when they operate between 30% and 90% of their full load design. When these energy per ton figures are linked to the typical hourly load profile, the chiller's arrival power consumption becomes more meaningful. This effect is called IPLV (integrated part load value) and governs the rating and testing of centrifugal and screw water chillers.

For an electric chiller, the IPLV in kW/ton on a weighted basis might be only 90.6% of the full-load valve of 0.70 kW/ton. In practice with a properly operated centrifugal or screw compressor chiller, this weighted power input will range from 85% to 91% of the full-load kW/ton. A similar factor may also apply to an absorption chiller's full-load (Btu per ton hour fuel energy input).

The full-load efficiency and IPLV of various types of chillers are given in Table 2.

Table 2 Chiller efficiency

Condenser type	*Compressor type*	*Minimum tons*	*Maximum tons*	*Rated kW/TR*	*Rated IPLV*	*Design CoP*
Water cooled	Scroll	1	80	0.79	0.78	–
		81	150	0.76	0.70	–
		151	300	0.72	0.70	–
		301	and up	0.64	0.61	–
Water cooled	Reciprocating	1	80	0.84	0.75	–
		81	and up	0.82	0.75	–
Water cooled	Gas engine	501	2000	–	–	1.80
Water cooled	Absorption (SE)	150	1000	–	–	0.65
	Absorption (DE)	150	1000	–	–	1.13
Water cooled	Centrifugal	1	150	0.62	0.62	–
		151	300	0.60	0.61	–
		301	and up	0.56	0.56	–
Air cooled	Scroll	1	80	1.25	1.10	–
Air cooled	Screw	1	and up	1.21	1.00	–
Air cooled	Reciprocating	1	and up	1.15	1.15	–

IPLV – integrated part load value; CoP – coefficient of performance

HVAC system types

Four-pipe fan coil

A four-pipe fan coil system utilizes units with a finned-tube coil, filter, and fan section. The fan recirculated air continuously from the space through the coil, which contains either hot or chilled water. Some units have electric resistance heater, steam or hot water coils. The filter is usually a cleanable or replaceable low efficiency (less than 25%) filter that protects the coil from clogging up with dirt and lint. Units can be connected to dampered openings in the outside wall to provide some outdoor air for ventilation. Fan coils are typically arranged in a floor-mounted configuration but horizontal overhead models are also available. Fan coils can be ducted to discharge through several outlets but fan static pressure capacity is usually very limited. Ventilation is usually provided by ducting tempered outside air to the return side of the unit or directly to the space.

Advantages

- System can economically provide many temperature control zones.

- The all-water systems conserve space and is useful where ceiling heights are restricted.
- Suitable for low water temperature heating such as solar or heat recovery.

Disadvantages

- Fans and motors are very inefficient.
- Dehumidification can be a problem where high latent loads are present.
- Fan coils are maintenance intensive and require regular filter replacement, fan and motor lubrication. Condensate drain pans are subject to clogging and overflowing.
- Fans can be noisy.

Common applications

- Used in apartment or hotel units
- Used in perimeter systems in office buildings
- Sometimes used in hospital patient rooms with very mixed results.

Package-terminal air-conditioners

Package-terminal air-conditioners are a class of equipment commercially available that usually heats and cools a single room application. The units are fully self-contained and consist of fan, filter, direct expansion cooling coil, compressor, air-cooled condenser coil, and condenser fan—all encased in an enclosure and made for through-the-wall applications. The units may be made into heat pumps by providing the proper refrigeration accessories. Heating from package-terminal air-conditioning units is usually from electric resistance coils. Units can be equipped with ventilation openings of limited size and capacity.

Advantages

- Low initial cost
- Very good for sub-metering
- Requires only small amount of space at perimeter of building

Disadvantages

- Poor dehumidification control
- No economizer operation
- Increased maintenance due to multiple units
- Condensate removal during periods of heavy condensation
- Limited ventilation control
- Can be very noisy

- Architectural impact on building envelope
- Can create drafts and poor air distribution
- Does not handle interior space
- Poor filtration

Common applications
- Hotels, apartments
- Small office buildings

Packaged rooftop single zone

A packaged rooftop single zone air-conditioning system utilizes unitary (factory fabricated) equipment. The single zone unit is fully self-contained and consists of a constant volume supply fan, direct expansion cooling coil, heating (when required) with gas furnace or electric resistance, filters, compressors, condenser coils, and condenser fans. Units can be equipped with outdoor air economizer sections that have no relief, barometric relief or power relief fans. Units can be purchased as heat pumps. Units are typically mounted on roof curbs but can be also mounted on structural supports or on grade. Package rooftop single zone units are typically controlled from a single space thermostat with one unit provided for each zone.

Advantages
- Low initial cost
- Compact arrangement uses no inside mechanical equipment space

Disadvantages
- Limited zoning capability
- Poor dehumidification control
- Limited capacity to handle ventilation air
- Higher maintenance costs

Common applications
- Small office buildings
- Schools
- Restaurants

Packaged single zone - split system

A packaged single zone split system utilizes unitary (factory fabricated) equipment that is fully self-contained to provide heating and cooling. The system consists of an indoor package and an outdoor package that are connected to each other with refrigerant piping and controls. The indoor package has a supply fan, filters, direct expansion cooling coil and heating (if required) with hot water, steam, gas furnace or electric resistance. The outdoor unit has compressor(s), condenser coil, and condenser fans. The indoor unit can be equipped with economizer but will not have any relief capability. Units can be purchased as heat pumps. Package single zone split systems are typically controlled from a single space thermostat with one unit provided for each zone.

Advantages

- Low initial cost but more than package rooftop systems
- Can be utilized where there is limited outside space available

Disadvantages

- Requires space inside the building
- Requires separate relief when economizers are used
- Limited distance between indoor and outdoor units
- Limited zoning capability
- Poor dehumidification control
- Limited capacity to handle ventilation air

Common applications

- Small office buildings
- Apartments
- Schools
- Restaurants

Packaged rooftop variable air volume

A packaged rooftop variable air volume air-conditioning system utilizes unitary (factory fabricated) equipment. The unit is fully self-contained and consists of a variable volume supply fan, direct expansion cooling coil, heating (when required) with gas furnace, hot water, steam or electric resistance, filters, compressors, condenser coils, and condenser fans. Units are usually equipped with outdoor air economizer sections that have no relief, barometric relief or power relief fans. Units are typically mounted on roof curbs but can be also mounted on structural supports or on grade. Air is distributed from the unit through ductwork (often medium pressure) to terminal units and then to the space through a low-pressure distribution system. The terminal units regulate the volume of air and often heat the air with hot water or electric resistance coils in response to space temperature conditions. Sometimes the terminal units are equipped with fans (fan-powered) to recirculate room air for energy conservation.

Sometimes a separate central hot air (hot deck) system is ducted to the terminal unit (double duct) where the cold and hot air streams are mixed and the volume of air is varied to satisfy the room temperature. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust fan.

Advantages

- Low initial cost for buildings that require multiple zones
- Compact arrangement uses no inside mechanical room space and very little shaft space
- Better dehumidification control than packaged single zones
- Energy efficiency of variable air volume

Disadvantages

- Can have impact on building aesthetics
- Higher maintenance than chilled water systems
- Limited capacity to handle ventilation air

Common applications

- Large office buildings
- Schools
- Retail or shopping malls
- Assembly halls or auditoriums
- Restaurants

Packaged variable air volume - split system

A packaged variable air volume split system utilizes a unitary (factory fabricated) equipment that provides heating and cooling. The system consists of an indoor package and an outdoor package that are connected to each other with refrigerant piping and controls. The indoor package has a variable air volume supply fan, filters, direct expansion cooling coil and heating (if required) with hot water, steam, gas furnace or electric resistance. The outdoor unit has compressor(s), condenser coil, and condenser fans. The indoor unit can be equipped with economizer but will not have any relief capability. Air is distributed from the unit through ductwork (often medium-pressure) to terminal units and then to the space through a low-pressure distribution system. The terminal units regulate the volume of air and often heat the air with hot water or electric resistance coils in response to space temperature conditions. Sometimes the terminal units are equipped with fans (fan-powered) to recirculate room air for energy conservation. Sometimes a separate central hot air (hot deck) system is ducted to the terminal unit (double duct) where the cold and hot air-streams are mixed and the volume of air is varied to satisfy the room temperature. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust fan.

Advantages
- Low initial cost but more than package rooftop systems
- Can be utilized where there is limited outside space available
- Good dehumidification control

Disadvantages
- Requires space inside the building
- Requires separate relief when economizers are used
- Limited distance between indoor and outdoor units
- Complicated control interface between inside and outside units

Common applications
- Large office buildings
- Schools
- Retail or shopping malls
- Assembly halls or auditoriums
- Restaurants

Water-cooled direct expansion, constant volume

The water-cooled direct expansion, constant volume system utilizes packaged self-contained water-cooled air-conditioning units. The units consist of a constant volume supply fan, direct expansion cooling coil, heating (if required) with hot water, steam, or electric resistance, filters (high efficiency available), compressors, and water-cooled condenser all housed within a sheet metal enclosure. The water-cooled condenser is hydronically connected to a two-pipe condenser water system. The condenser water system rejects heat from a cooling tower that is often a closed circuit fluid cooler. The unit can be equipped with an airside or waterside economizer. A constant volume of air is distributed from the unit to the space through a low-pressure distribution system. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust (spill) fan. This system is typically controlled from a single space thermostat with one unit provided for each zone.

Advantages

- Eliminates need for central chilled water plant
- Does not require as much floor space or shaft space compared to central systems
- Units can operate at customized time schedules for greater flexibility
- More units means less impact to tenants should an air handler fail
- Smaller units means less static pressure at each fan
- Excellent for sub-metering
- Water cooled condensing is more energy efficient than air cooled

Disadvantages

- Airside economizers may be more difficult to integrate into building architecture
- Smaller fans are less efficient
- Unit sizing cannot take advantage of building wide diversity
- More units means more maintenance in generally tighter mechanical space
- May cost more than centralized systems
- Limited capacity to handle ventilation air

Common applications

- Small offices
- Retail or shopping malls
- Schools
- Restaurants

Water-cooled direct expansion, variable air volume

A water-cooled direct expansion, variable air volume system utilizes packaged self-contained water-cooled air-conditioning units. The units consist of a variable air volume supply fan, direct expansion cooling coil, heating (if required) with hot water, steam, or electric resistance, filters (high efficiency available), compressors, and water-cooled condenser all housed within a sheet metal enclosure. The water-cooled condenser is hydronically connected to a two-pipe condenser water system. The condenser water system rejects heat from a cooling tower that is often a closed circuit fluid cooler. The unit can be equipped with an airside or waterside economizer. Air is distributed from the unit through ductwork (often medium-pressure) to terminal units and then to the space through a low pressure distribution system. The terminal units regulate the volume of air and often heat the air with hot water or electric resistance coils in response to space temperature conditions. Sometimes the terminal units are equipped with fans (fan-powered) to recirculate room air for energy conservation. Sometimes a separate central hot air (hot deck) system is ducted to the terminal unit (double duct) where the cold and hot air streams are mixed and the volume of air is varied to satisfy the room temperature. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust fan.

Advantages

- Eliminates need for central chilled water plant
- Does not require as much floor space or shaft space compared to central systems
- Units can operate at customized time schedules for greater flexibility
- More units means less impact to zones should an air handler fail
- Smaller units means less static pressure at each fan
- Excellent for sub-metering
- Water-cooled condensing is more energy efficient than air cooled

Disadvantages

- Airside economizers may be more difficult to integrate into building architecture
- Smaller fans are less efficient and may not use the most efficient volume control
- Unit sizing cannot take advantage of building wide diversity
- More units means more maintenance in generally tighter mechanical space
- May cost more than centralized systems
- Limited capacity to handle ventilation air

Common applications

- Large office buildings
- Schools
- Retail or shopping malls
- Assembly halls or auditoriums
- Restaurants

Chilled-water, constant volume

A chilled-water, constant volume system utilizes a constant volume air-handling unit to provide heating and cooling to a single zone. The unit consists of a constant volume supply fan, chilled water cooling coil, hot water, steam or electric heating coil, filters (can be medium or high efficiency) and a mixing plenum where outdoor air and return air are mixed. The unit can utilize an airside or waterside economizer. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust fan. Chilled water constant volume systems are typically controlled from a single space thermostat with one unit provided for each zone.

Advantages
- Simplicity of unit and control
- Low maintenance

Disadvantages
- Hard to sub-meter in multiple zones
- Requires mechanical space
- Dehumidification can be a problem where high latent loads are present.

Common applications
- Retail or shopping malls
- Assembly halls or auditoriums
- Laboratories
- Schools

Chilled water variable air volume - central

A central chiller plant supplies chilled water to one or more central variable air volume air-handling units. The variable air volume units consist of mixing plenums where outdoor air and return air are mixed, filters (can be medium or high efficiency), cooling and/or heating coils, and fans, all contained in an insulated sheet metal housing. Air is distributed from the air handlers through ductwork (often medium-pressure) to terminal units and then to the space through a low-pressure distribution system. The terminal units regulate the volume of air and often heat the air with hot water or electric resistance coils in response to space temperature conditions. Sometimes the terminal units are equipped with fans (fan-powered) to recirculate room air for energy conservation. Sometimes a separate central hot air (hot deck) system is ducted to the terminal unit (double duct) where the cold and hot air streams are mixed and the volume of air is varied to satisfy the room temperature. Air is returned from the space to the unit for recirculation or exhaust. Return/exhaust fans can be used for this purpose or the supply fan can be used with assistance from a power exhaust fan.

Advantages

- Plant equipment is centralized for ease of maintenance
- Central equipment can take advantage of load diversity for optimal sizing
- Can use airside economizers effectively
- Variable speed drives for fan volume control are more cost-effective
- Provides a great deal of flexibility for multiple zones
- Provides good dehumidification control
- Well suited for good control of building pressurization
- Good control of ventilation air quantities
- Opportunity for higher levels of filtration

Disadvantages

- Can be more expensive than packaged equipment
- Hard to sub-meter in multiple zones
- Requires mechanical space, i.e. equipment rooms, shafts, etc.
- May require more duct space that may increase the height of the building
- Special attention to acoustics is required

Common applications

- Large office buildings
- Large retail buildings
- Institutional buildings

Chilled water variable air volume - distributed

A chilled water variable air volume - distributed system type is similar to the central variable volume approach except that the air-handling units are decentralized and distributed throughout the building. The air handlers consist of mixing plenums, filters, cooling and/or heating coils and supply fans. Air is distributed to terminal units that vary the volume and sometimes heat the air with hot water or electric resistance coils to maintain space temperature. Fan-powered terminal units or double-duct terminal units are also viable. Typically, the supply fan is used to return the air and is assisted by a power exhaust fan when airside economizers are used.

Advantages

- Does not require as much floor space or shaft space as compared to central systems
- Units can operate at customized time schedules for greater flexibility
- More units means less impact to zones should an air handler fail
- Smaller units means less static pressure at each fan

Disadvantages

- Airside economizers may be more difficult to integrate into building architecture
- Smaller fans are less efficient and may not use the most efficient volume control
- Unit sizing cannot take advantage of building wide diversity
- More units means more maintenance in generally tighter mechanical space
- May cost more than centralized systems

Common applications

- Large office buildings
- Schools
- Hospitals
- Hotels

Water loop heat pump

A water loop heat pump system utilizes water-to-air heat pumps that are connected hydronically with a common two-pipe condenser water system. Water-to-air heat pumps have a self-contained refrigeration system that includes a compressor; water-cooled condenser, and direct expansion cooling coil. These components are housed within a cabinet that also contains a fan and filters. In a cooling mode, the refrigeration system cools the air and rejects the heat into the condenser water loop. The units are equipped with a reversing valve that allows them to act as a heat pump. During the heating mode, the refrigeration system extracts heat from the condenser water loop and rejects it into the space via the direct expansion coil. The common condenser water loop is connected to a cooling tower (usually a closed circuit fluid cooler) where excess heat is rejected to the atmosphere. The loop is also connected to a heating device such as a boiler (fossil fuel or electric resistance) where heat can be added. The system is inherently a heat recovery system, because the heat that is rejected from interior zones can be redirected to perimeter zones in a heating mode. Ventilation is provided by a separate air-handling system that provides tempered outdoor air to each heat pump or sometimes into the ceiling plenum.

Advantages

- Easy to sub-meter multiple zones
- Little or no mechanical room or shaft space required
- Provides heat recovery
- Uses low water temperatures for heating making it attractive for solar or ground source or thermal storage

- Entire system is not shut down by failure of a unit
- Units can be installed as space is leased or occupied

Disadvantages

- Maintenance access is restricted, particularly for concealed units above ceilings
- No opportunity for economizer operation
- Lack of dehumidification control
- Expensive to add zones, not very flexible to accommodate changes
- Poor quality filtration
- System must be winterized (usually glycol) in climates that are below freezing
- Noise from cycling compressors can be annoying
- Condensate removal, especially when above ceilings

Common applications

- Office buildings
- Retail or shopping malls
- Hotels and apartments
- Schools

Variable volume variable temperature

A variable volume variable temperature system utilizes a packaged rooftop single zone unit in a configuration that provides some ability for additional temperature control zones. Dampers are placed in the low-pressure distribution ductwork downstream of the unit. Space thermostats modulate the dampers to maintain temperature. A master controller monitors each zone thermostat and controls the unit to provide either heating or cooling depending on the priorities established. A damper bypasses air from the supply duct to the return duct to maintain a near constant volume from the fan regardless of the zone damper position. To be effective, the temperature control zones should be thermally similar (for example, all interior zones) to avoid rapid switching from heating to cooling that can be typical with this type of system.

Advantages

- Provides additional temperature control zones at low initial cost
- Compact arrangement uses no inside mechanical equipment space

Disadvantages

- Rapid switching between dissimilar zones can lead to energy waste

- Complex controls add maintenance costs
- Can inhibit ventilation to zones where the air volume is decreased

Common applications
- Small offices
- Retail or shopping malls

Unit ventilator

A unit ventilator can be mounted in a vertical or horizontal position. These units are used to provide heating, cooling, and ventilation for a single space. The units consist of constant volume fans, chilled water or direct expansion cooling coils, hot water, steam, or electric resistance heating coils, filters (low efficiency 25% or less), and outdoor and return air dampers all enclosed in a heavy gage metal housing. Ventilation and/or economizer air is drawn from adjacent openings in the outside wall. Relief is either by gravity or powered exhaust remote from the unit.

Advantages
- Low initial cost
- Less mechanical space

Disadvantages
- Poor filtration
- Poor dehumidification control
- Poor air distribution, subject to drafts

Common applications
- Schools

All the above-mentioned systems have their specific application depending on the building type, floor area and availability of roof (Table 3).

Table 3 Recommended system type

Building	*Minimum floors*	*Maximum floors*	*Minimum area* (sq. feet)	*Maximum area* (sq. feet)	*Roof availed*	*Rank*	*System type*
Office	1	2	1	20 000	Yes	1	Packaged rooftop single zone
						2	Variable volume variable temperature
	1	2	1	20 000	No	1	Packaged rooftop - split system

Contd...

Table 3 *Contd...*

Building	*Minimum floors*	*Maximum floors*	*Minimum area (sq. feet)*	*Maximum area (sq. feet)*	*Roof availed*	*Rank*	*System type*
	1	2	20 001	10 000 000	Yes	1	Packaged rooftop VAV
						2	Variable volume variable temperature
						3	Water loop heat pump
						4	Packaged roof top single zone
	1	2	20 001	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled DX, VAV
						4	Water loop heat pump
	3	6	1	180 000	Yes	1	Packaged rooftop VAV
						2	Chilled water VAV - central
						3	Chilled water VAV - distributed
						4	Water cooled DX, VAV
						5	Water loop heat pump
	3	6	1	180 000	No	1	Chilled water VAV- central
						2	Chilled water VAV - distributed
						3	Water cooled DX, VAV
						4	Water loop heat pump
	3	6	180 001	1 000 000	No	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled DX, VAV
						4	Water loop heat pump
	3	6	180 001	10 000 000	Yes	1	Packaged rooftop VAV
						2	Chilled water VAV - central
						3	Chilled water VAV - distributed
						4	Water cooled DX, VAV
						5	Water loop heat pump
	7	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled DX, VAV
						4	Water loop heat pump
	7	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water - distributed
						3	Water cooled DX, VAV
						4	Water loop heat pump
Retail	1	1	1	10 000 000	Yes	1	Packaged rooftop single zone
	1	1	1	20 000	No	1	Packaged single zone - split system
	1	1	20 001	10 000 000	No	1	Water loop heat pump
						2	Chilled water - constant volume
						3	Water cooled DX, constant volume
	2	6	1	10 000 000	Yes	1	Packaged rooftop VAV
						2	Chilled water VAV - central
						3	Water loop heat pump
	7	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Water loop heat pump

Contd...

Table 3 *Contd...*

Building	*Minimum floors*	*Maximum floors*	*Minimum area (sq. feet)*	*Maximum area (sq. feet)*	*Roof availed*	*Rank*	*System type*
						3	Chilled water - constant volume
						4	Water cooled DX, constant volume
	2	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Water loop heat pump
						3	Chilled water - constant volume
						4	Water cooled DX, constant volume
Assembly < 300 people	1	1	1	10 000 000	Yes	1	Packaged rooftop single zone
						2	Packaged rooftop VAV
	1	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water - constant volume
Assembly >300 people	1	1	1	10 000 000	Yes	1	Chilled water - constant volume
						2	Water cooled DX, constant volume
	1	200	2	10 000 000	No	1	Chilled water - constant volume
						2	Water cooled DX, constant volume
	2	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Chilled water VAV - constant volume
						3	Chilled water -constant volume
						4	Water cooled DX, constant volume
Hotel/ Motel	1	3	1	10 000 000	Yes	1	Package terminal air-conditioners
	4	200	1	10 000 000	Yes	1	Four pipe fan coil
						2	Package terminal air-conditioners
						3	Water loop heat pump
	1	3	1	10 000 000	No	1	Packaged terminal air-conditioners
	4	200	1	10 000 000	No	1	Four pipe fan coil
						2	Package terminal air-conditioners
						3	Water loop heat pump
School	1	2	1	160 000	Yes	1	Packaged rooftop VAV
						2	Chilled water VAV - central
						3	Water cooled DX, VAV
						4	Unit ventilator
	1	2	1	160 000	No	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Packaged rooftop VAV
						4	Water cooled DX, VAV
	1	2	160 001	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled - DX, VAV
	3	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled DX, DX, VAV
	3	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water VAV - distributed
						3	Water cooled DX, VAV

Contd...

Table 3 *Contd...*

Building	*Minimum floors*	*Maximum floors*	*Minimum area (sq. feet)*	*Maximum area (sq. feet)*	*Roof avai-led*	*Rank*	*System type*
Restaurant	1	1	1	20 000	Yes	1	Packaged rooftop single zone
	1	1	1	20 000	No	1	Packaged single zone - split system
	1	200	20 001	10 000 000	Yes	1	Packaged rooftop VAV
						2	Packaged rooftop single zone
	1	200	20 001	10 000 000	No	1	Water cooled DX, VAV
						2	Packaged VAV, split system
						3	Packaged VAV - split system
Multi-family	1	3	1	10 000 000	Yes	1	Package terminal air-conditioners
						2	Packaged single zone - split system
	1	3	1	10 000 000	No	1	Package terminal air-conditioners
						2	Packaged single zone - split system
	4	200	1	10 000 000	Yes	1	Package terminal air-conditioners
						2	Four pipe fan coil
						3	Water loop heat pump
	4	200	1	10 000 000	No	1	Package terminal air-conditioners
						2	Four pipe fan coil
						3	Water loop heat pump
Laboratory	1	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Chilled water - constant volume
Health Care	1	200	1	10 000 000	Yes	1	Chilled water VAV - central
						2	Chilled water - constant volume
						3	Four pipe fan coil
Warehouse	1	200	1	10 000 000	Yes	1	Direct fired make-up air
						2	Gas or HW unit heaters
Laboratory	1	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water - constant volume
Health Care	1	200	1	10 000 000	No	1	Chilled water VAV - central
						2	Chilled water - constant volume
						3	Four pipe fan coil
Warehouse	1	200	1	10 000 000	No	1	Direct fired make-up air
						2	Gas or HW unit heaters

VAV - variable air volume; DX - direct expansion

Measurements/ field testing

The performance of a given system depends upon the performance of the equipment involved. The parameters needed to be looked into are

- inside and outside design conditions,
- measured flows and capacities of all the equipment used in the system,
- comparison of the measured and design capacities, and
- comparison of energy consumption with the design values.

To measure and evaluate capacities, the following instruments and process will be required for different equipment.

Air flow The air flow is calculated by measuring velocity across a fixed opening having a definite area. The velocity is determined by either measuring the pressure drop across the opening with the help of a pitot tube and manometer or a rotating vane/hot-wire anemometer.

Water flow The water flow in a piping system is measured by

- taking the pumping head at the pump discharge and then referring the pump characteristic curves,
- measuring the pressure drop across a fixed length of piping and then using equations,
- taking the pressure drop across the heat exchangers and then comparing with the design pressure drop across the heat exchanger, and
- measuring the flow with a portable non-intrusive ultrasonic flow meter.

Rotation/speed These can be measured with the help of a stroboscope, a non-contact type electronic techometer, and odometer

Temperature/humidity Two types of temperatures, dry bulb and wet bulb, are measured by

- sling psychrometers and
- electronic thermohygrometers.

Electrical parameters Electrical measurements are taken with the help of the clamp-on type portable power masters, which can measure voltage, current, kilowatt, power factor, and frequency.

Performance evaluation

An air-conditioning and refrigeration plant is efficient when all the system components, i.e., the compressor, the condenser, the evaporator, and the condenser cooling (heat rejection) system are working in matched conditions. This means that under peak operating conditions they must perform to their optimum output.

To evaluate the performance, the testing engineer must have the parameters and duty requirements of each component. The engineer must also have the equipment capacity charts and tables. The test readings for each equipment have also been discussed. The next section discusses the method by which individual equipment capacities may be calculated from the test readings.

Compressor From the operating suction and the discharge pressure, the capacity may be evaluated from the rating charts. The operating kilowatt input can give the value of input power. This can be then compared with the rated input.

Condenser Water or air-flow rate and temperature rise across the condenser gives the heat rejection. The flow rates and temperature rise should be compared with the design parameters. Any variation must be analysed.

The equation for water-cooled condenser is as follows.

Heat rejected (Btu/hour) = 500 × US gallon per minute × ΔT (°F)
or
Heat rejected (kW) = 4.182 × litre/second × ΔT °C

Condenser water flow is measured in US gallon per minute in FPS system and litre per second in SI units

ΔT is the difference of leaving condenser water temperature and entering condenser water temperature

The equation for air-cooled condenser is as follows.

Heat rejected (Btu/hour) = 1.08 × CFM (cubic feet/minute) × ΔT °F
or
Heat rejected (kW) = 1.20 × litre/second × ΔT °C

The air flow in condenser is measured in CFM in FPS system and litre per second in SI units.

The heat rejected must be compared with the rated heat rejection as per the rating chart. For each operating condition of the compressor the heat rejected will vary.

Chiller or evaporator Water or brine flow rate and temperature drop across the chiller gives the cooling load. The flow rates and temperature drop should be compared with the design parameters. Any variation must be analysed.

The equation for water/brine is as follows.

Cooling produced (tons) = [GPM (US) × specific heat × specific gravity × ΔT °F]/24
or
Cooling produced (kW) = 4.182 × litre/second × specific heat × specific gravity × ΔT °C

Cooling coil

Air-flow rate and enthalpy drop across the cooling coil will give the cooling load. The flow rates and enthalpy drop should be compared with the design parameters and any variation analysed.

The equation for cooling coil is as follows.

Cooling capacity of coil (Btu/hour) = 4.5 × CFM × ΔH
or
Cooling capacity of coil (kW) = 1.2 × litre/second × ΔH

ΔH is the change in enthalpy of return air and supply air and measured in Btu per pound in FPS unit and kilo joule per kilogram in SI units.

Cooling tower

The heat rejected at the condenser must match the heat rejected at the cooling tower. The equation for cooling tower is the same as that of the water-cooled condenser.

For the cooling tower, the approach must also be evaluated. The approach is the difference between ambient wet bulb temperature and the water outlet (sump) temperature.

The efficiency of the cooling tower is

$$\frac{\text{Entering water temperature} - \text{leaving water temperature}}{\text{Entering water temperature} - \text{ambient wet bulb temperature}}$$

The efficiency thus obtained must be compared with the rated efficiency.

Energy conservation opportunities

Installation of variable speed drives

At air-handling units in fan motors

One of the most useful electrical developments in recent years for motor speed control is the AC VSD (variable speed drive). These new drives typically use electronic circuitry to vary the output frequency, which, in turn, varies the speed of the motor. Since the power required driving a centrifugal fan or centrifugal pump is proportional to the cube of the fan or pump speed, large reductions in power consumption are obtained at reduced speed. A quality VSD usually obtains greater savings than does a variable-pitch inlet vane or other mechanical flow volume control.

As the load on different AHUs (air-handling units), which depends on occupancy, lighting, and ambient conditions, keeps changing throughout the day and during the year, the required CFM (air flow) for producing the required amount of cooling also keeps changing. Since the AHUs are designed for peak conditions, it is very conveniently assumed that the annual average required CFM could be 80% of the operating CFM. If the VSD is installed at different AHUs, the daily and seasonal variation in the required CFM would be possible and energy could be saved.

At cooling tower fan motors
Since the cooling towers are generally designed for the worst wet bulb temperature and peak load conditions, the air flow generated by fans is the maximum flow required for adequate cooling of the condenser water. The condition of air entering cooling towers varies depending on the season and time of the day. It means the operating conditions keep changing throughout the year and even at low loads and lower ambient wet bulb temperatures, the fan with a single-speed motor operates at constant speed, which leads to wastage of energy. In order to utilize the motor drive power effectively, the fan output must be optimally adapted to the cooling requirement and cooling effect, which is governed by climatic conditions.

At secondary chilled water pumps
Secondary pumps in a primary/secondary-chilled water pumping system are used to distribute the chilled water to the loads from the primary production loop. The primary/secondary pumping system is used to hydronically decouple one piping loop from another. The primary pump is used to maintain a constant flow through the chillers while allowing the secondary pumps to vary in flow, increase control, and save energy.

While the primary/secondary system with two-way valves improves energy savings and eases system control problems, the true energy savings and control potential are realized by adding VSDs.

Low leakage dampers

Allowing unwanted outdoor air into a building during periods when the outdoor air damper should be closed can unnecessarily add to the load of the mechanical system.

This situation can occur during those periods when the outdoor air damper is supposed to be closed, such as unoccupied periods, or where minimum air is controlled by a separate damper and the primary damper used to maintain mixed air temperature leaks. It will also occur during occupied hours in those locations, which allows the outdoor air dampers, to close when the outdoor air drops below some prescribed temperature.

The use of low leakage dampers, which restrict leakage to less than one per cent, will reduce this wasteful load. Standard dampers can allow 5% to 30% leakage when closed.

Reduced minimum outdoor air

Reducing the amount of outdoor air for ventilation can reduce the amount of mechanical heating and cooling required for tempering air for space conditioning. However, minimum outdoor air has to be maintained, as per latest ASHRAE's standard. Depending upon occupancy, outdoor air can be adjusted.

This reduction can be used on any system that provides minimum fresh air to a building. By adjusting the damper linkage or the minimum positioning switch, the amount of outdoor air can be reduced to the minimum required by state or local codes.

Careful measurements are required to relate damper blade position to outdoor airflow.

Unoccupied ventilation reduction

Energy savings are possible by closing off outside ventilation air during unoccupied periods.

Required ventilation for conditioned spaces need only be supplied while they are occupied. Many fan systems are required to run during unoccupied periods simply to maintain night temperatures, but are controlled so that there is no way to reduce the amount of outside air into the space during unoccupied periods. Almost all systems have to be started before occupancy in order to warm up or cool down the space to comfort conditions. By the addition of a time clock or automation system the outside air can be shut off during these unoccupied periods. The result is a lower demand on HVAC (heating, ventilation, and air-conditioning) equipment to heat or cool that unnecessary outside air.

Enthalpy control/dry bulb economizer

Using outside air to cool a building can result in lower mechanical refrigeration costs whenever the outdoor air has a lower total heat content (enthalpy) than the return air. This can be accomplished by an 'integrated economizer' or enthalpy control.

Dry bulb economizer

The operation of the 'integrated economizer' can be made automatic by providing dampers capable of providing 100% outdoor air with local controls that sequence the chilled water or DX (direct expansion) coil and dampers. During economizer operation, when discharge (or space) temperature rises, the outdoor damper opens first; then on a further rise, the cooling coil is turned 'on'.

Economizer operation is activated by outside air temperature, say 72 °F dry bulb temperature. If the outside air is below 72 °F, the above described economizer sequence occurs. Above 72 °F, outside air-cooling is not economical, and the outdoor air damper closes to its minimum position to satisfy ventilation requirement only.

Enthalpy control

If an economizer system is equipped with an enthalpy control, savings will accrue due to a more accurate change-over point. The load on a cooling coil for an air-handling system is a function of the total heat of air entering the coil. Total heat is a function of two measurements, DBT (dry bulb temperature), and RH (relative humidity), or dew point. The enthalpy control measures both conditions (DBT and RH), in the return air duct and outdoors. It then computes which air source would impose the lowest load on the cooling system. If outside air is the smallest load, the controller enables the economizer cycle (DBT economizer control savings will be less than enthalpy control, except in dry climates).

Exhaust air control

Exhaust air fans frequently operate more often than required to perform their purpose. By controlling these fans so that they function only when needed, appreciable electrical energy may be saved. Exhaust air fans have two major functions: removal of odours or fumes and excess heat build-up.

When controlling exhaust air fans that remove objectionable fumes or odours, their operation must be scheduled either automatically or manually, so that they operate only when the objectionable air contaminants are being generated. Examples of this type of exhaust air fans are laboratory hoods, kitchen vents, and rest-room vents.

These exhaust fans used to control excess heat build-up in spaces (i.e., storerooms, warehouses, and garages), must be placed on a thermostatic control.

In some applications, the exhaust air fan can be interlocked with the supply air fan serving the same area to accomplish the operations described above. Installation of backdraft dampers in the exhaust air fan discharge will prevent uncontrolled outside air from entering the conditioned space or prevent the loss of conditioned air from the space. This depends upon the relative static pressure within the building when the exhaust fan is off.

Retrofit of central fans for variable air volume usage

Conversion of constant volume of air distribution systems to a VAV (variable air volume) can represent energy savings. It often requires that some action be taken to prevent the central fans from operating in critical areas and producing excessive negative statistics in the building.

More importantly, the retrofit of the central fan system of a VAV system results in considerable energy savings in fan horsepower. Only those fans, which have forward-curved impellers or are equipped with some means of flow adjusters, such as vortex vanes are discussed below.

Fans with air foil or backward curved impellers without vortex vanes require additional of vanes or some other means of volume control, such as an eddy current clutch or variable speed drive, before the advantage of the VAV operation can be realized. This requires an on-site evaluation to select the least costly approach.

Forward-curved fan systems

Most typical of rooftop and packaged fan systems, the supply fan needs no retrofit since the throttling action of the VAV terminal units automatically results in reduction of fan horsepower, and energy consumed.

The return fan may require the addition of a damper in the return duct system if building statics become objectionable. This damper would be controlled by a space static sensor referenced to outdoors.

Fans with vortex vanes

Fans with vortex vanes require a different approach. The throttling effect of the terminal units, although producing some energy savings, may push the fan operation into the area not recommended by the fan manufacturer. Therefore, static pressure control should be installed on both the supply and return fans.

The supply fan static control maintains the minimum supply duct static required delivering the minimum required CFM.

The return fan static control senses changes in space pressure relative to outdoors and maintains the space at a balanced or slightly positive value to minimize infiltration.

A VAV system delivers air in quantities based on the actual instantaneous load requirements. The more variable the building load, the greater the savings by converting to VAV. In a general office building, the difference in fan CFM between a constant volume system and the average flow in a VAV system is typically 30%–40%. This results in electrical savings in the fan horsepower required.

Conversion of dual duct constant volume to variable air volume

Dual duct constant volume systems continually mix air from the hot and cold ducts to achieve the temperature that satisfies the load. (Exceptions are the hottest or coldest days when the design conditions are occurring and only hot or cold air is provided.)

This mixing is wasteful, since it mixes cooled air with heated air and energy is consumed in both the processes.

Space temperature can be better maintained by reducing the air volume instead of its temperature.

The interior zones of practically all-dual duct constant volume systems have hot air supplied solely to temper the cold air being delivered during off-peak operation. By converting these to single-duct variable volume systems, the need for hot air supply in these areas is totally eliminated, along with the wasteful mixing.

Recycling of cool air effect in air-conditioned spaces

The availability of air-to-air heat exchangers namely, air-to-air plate heat exchangers, heat pipes, and heat wheels have opened up possibilities for saving energy in air-conditioning. These are discussed below.

Air-to-air plate heat exchangers

This heat exchanger has a series of plates to provide sufficient area to facilitate heat transfer for low temperature differences. This device is now innovatively used for energy saving in low relative humidity air-conditioning.

Heat pipes

Heat pipes devices usually consist of sealed finned tubes with a wick lining on the inner side. The tube contains a working fluid, which evaporates from the hot end of the tube and condenses at the cold end, thus transferring heat. The working fluid is returned to the hot end by capillary action of the wick. The heat conductivity of heat pipes is about 1000 times that of copper and the system can operate with very low temperature.

Heat wheel

Heat wheels are rotary heat exchangers packed with aluminium honeycomb fill. One half of the wheel is in contact with the warm air and the other half with cold air. The sensible heat of the incoming warm air is given to the aluminium and the wheel rotates slowly releasing the heat to the exhaust cold air during the second half of the revolution. Desiccant coated heat wheels are also available where, in addition to sensible heat transfer, latent heat transfer also takes place as the desiccant absorbs moisture in one half of the revolution and releases it in the second half of the revolution.

Energy conservation methods

Heating

Reducing the building heating load provides a major opportunity for energy conservation in cold climate. Some of the ways of achieving this are listed below.

Design criteria

- Use ample insulation throughout the building. Inadequate insulation while constructing the building would result in considerable loss of energy.

- Prefer using 99% winter outdoor design temperature level wherever possible instead of 99.6% winter outdoor design temperatures.
- Use inside design temperatures that provide comfort but not excessive temperature. The practice of using temperatures as high as 75 °F is unnecessary. Consider using 68 °F or 70 °F.
- Be certain all windows and doors are weather stripped and have double-glazed windows with low infiltration.
- Heat losses shall be calculated using thorough, correct procedures.
- The building architectural design (orientation, use of glass, type of materials, etc.) should be consistent with reducing energy consumption.

Cooling

- Use high R-valve insulation throughout the building.
- Use the 1.0% summer outdoor design dry bulb temperature and simultaneously wet bulb temperature instead of 0.4%.
- Use inside design dry bulb temperature of 75–80 °F. These provide adequate comfort for most application. Past practice of designing for 75 °F or even lower is wasteful.
- Minimize use of glass in building unless used on the south side for receiving solar heat in the winter. Consider use of double-glazed low-e windows with effective interior shading devices.
- Consider outside construction features (louvers and fins), which provide shading of glass.
- Orient the building so that solar radiation in summer is minimum on sides with large glass areas.
- Avoid excessive lighting levels. Use efficient lighting devices that convert electrical energy into light. A 15 W/m^2 should be considered as the lighting load instead of 20 W/m^2.
- Above all, use proper calculation procedures that account for heat storage and time lag.

System design

- Fans should be chosen for close to maximum efficiency, particularly with high energy with high energy costs. This falls in the middle range of the pressure–CFM curve. Avoid the temptation of selecting a fan far out on the CFM curve, near maximum delivery. This temperature is great, because it may mean a smaller fan and, therefore, lower initial cost, but the operating cost will be high.
- Fans should not be selected to the left of the peak pressure on the fan curve. At these conditions, the systems operation

may be unstable; there may be pressure fluctuation and excess noise generated.

- Fans may have pressure curve of varying steepness. If it is expected, that there will be considerable changes in system resistance (kitchen exhaust with masonry ducts), but constant CFM is required, a fan with a steep curve is desirable. For VAV systems, a flat-curve type is desirable.
- Use air foil blade centrifugal fans, which have the highest efficiency.
- Do no allow extra pressure loss as a 'Safety factor' in the duct systems.
- If volume control of the fan is to be used, inlet guide vane dampers are preferable to outlet dampers, less power will be used when the volume flow rate is reduced.
- Reducing fan speed to reduce flow is the most efficient method for reducing power consumption. Use variable speed drives in fans so that the volume flow rate is reduced.
- Reducing fan speed to reducing flow rate is the most efficient method for reduced power consumption. Use variable speed drives in fans for significant energy savings.

Pumps

- A pump that is operating near the point of maximum efficiency should be selected. This generally falls in the mid-range of pump flow capacity.
- For hydronic systems, it is preferable to select a pump operating at 1450 rpm rather than 2900 rpm. At 2900 rpm, a smaller pump can be used, but the higher speed results in higher noise levels that may be disturbing in occupied areas.
- It is not advisable to select a pump operating neat its maximum capacity, even through a smaller pump results from this choice. Select a pump that results from this choice. Select a pump in the vicinity of 50%–75% of maximum flow.
- The steepness of the head-flow curves varies among centrifugal pumps, depending on their design. It is recommended that pumps with flat head characteristics curves be used for hydronic systems (CHW systems). If there is a large change in flow rate, there will be a corresponding small change in the pump's head. This makes balancing and controlling flow rates easier. A steep head curve pump may be used in a system where the systems pressure resistance is expected to gradually increase with time, yet where it is desired to maintain reasonably constant flow rate as in the cooling water circuit.

Refrigeration

- Use refrigeration compressors that reduce power requirements as load decreases. For reciprocating compressors, this would involve use of cylinder unloaders. For centrifugal compressors, this would involve use of speed control or inlet guide vanes.
- Select and operate equipment with highest evaporating (or chilled water) temperature and lowest condensing temperature consistent with maintaining satisfactory space conditions and satisfactory equipment performance.
- Use condenser heat for heating needs by recovery heat.
- Use multiple equipment on larger projects so that each operates close to full load more often.
- Use some form of total energy systems such as gas engines/ turbines or steam with centrifugal absorption machines.

Air-handlers

- Use a VAV-type air-conditioning system, if suitable. This is generally the most energy-efficient system, because the reduction in air quality directly lower power use.
- Avoid terminal reheat systems, which cool and then reheat air, unless reheating is done through energy from waste.
- Dual-duct and multi-zone systems or any other types that mix hot and cold air (or water) are inherently energy wasteful, and they should be avoided.
- Try to avoid using preheat coils for outside air. This often results in heating and re-cooling, a ridiculous waste of energy. Instead, design the intake plenum so that the outside and return air mix thoroughly.

Controls

- Use enthalpy control for the supply air. Enthalpy control devices sense and measure the total enthalpy of outside and return air, and adjust the dampers to provide the air mixture proportion that will provide the most economical natural cooling, thus reducing or eliminating need for refrigeration at times.
- Use automatic time switching to start and stop equipment according to need.
- Use night and weekend automatic temperature set-back for unoccupied spaces.
- Investigate the possible use of a centralized computer control systems (building automation system) designed to provide the most efficient operation at all times.

Operation and maintenance

- Retest and rebalance the systems completely at regular scheduled intervals during its lifetime. Often the initial balancing procedures are all a system ever receives, and it gradually loses efficiency.

- Shut down any equipment when not needed. The use of the space must be examined before this decision is made.
- Set back temperatures when spaces are not occupied. In winter, the temperature can often set back to 55 °F.
- Start equipment shut down or temperature set back shortly before occupants leave.
- Start equipment as late as possible before occupants arrive consistent with achieving comfort.
- Utilize natural pre-cooling at night from outdoor air (enthalpy control does this automatically).
- Close outdoor air dampers near the beginning and end of operation each day when equipment is providing heating or cooling.
- Turn-off unneeded lighting.
- Check that solar shading devices are used properly.
- Check and replace or clean air filters when resistance reaches design conditions.
- Clean all heat transfer surfaces regularly (coils, tubes).
- Set chilled water temperature (high) and condensing water temperatures (low) consistent with comfort and design values. Condensing water temperatures must not be so low that the refrigeration equipment malfunctions.
- Adjust burners and draft on furnaces and boilers periodically to ensure complete combustion and minimum excess air.
- Arrange for reliable water treatment services from a specialist for boiler and condensing water systems. Dirt in the systems will reduce heat transfer as well as harm equipment.
- Check operation of air-vents regularly.
- Limit the maximum electric power demand of the systems. This is accomplished by starting each piece of equipment in sequence, with ample time in between. In addition some equipment can be furnished with devices that limit their power demand.
- Check duct work regularly for leaks that may develop
- Check outside windows and doors regularly for abnormal cracks.
- Perform routine maintenance regularly (lubrication, belt tension, inspection for damage or breakage).
- Check that objects (books, clothing, furniture) are not obstructing air flow through room terminal units.
- Check that air distribution outlets have not been obstructed or tampered with.
- Check that room thermostats or humidistats have not been readjusted by unauthorized personnel. Provide locks, if necessary.

Case studies

Case study 1

Replace inefficient chilling plant with new efficient plant in a 100-room luxury hotel.

Plant type : Reciprocating
Plant capacity : 100 TR
Rated compressor motor : 90 kW
Rated full load efficiency : 0.9 kW/TR
Average operating efficiency of plant : 1.024 kW/TR
Average operating load on compressors : 67 kW
Total operating hours/year : 6998 hours
Total energy being consumed by plant : 6998 × 67 = 468 866 kWh
Average cooling being produced by plant in a year : 67 × 1.024 = 68.6 TR
Total cooling being produced by plant in a year : 6998 × 68.6 = 480 062 TR
Rated efficiency of new plant : 0.7 kW/TR (screw chiller)
Projected energy consumption by new plant : 480 062 – 0.7 = 336 043 kWh
Total annual energy saving : 468 866 – 336 043 = 132 823 kWh
Total savings in energy cost (@ Rs 5/kWh) : 132 823 × 5 = Rs 664 115
Initial investment required : Rs 2 000 000
Simple payback period : $\frac{2\,000\,000}{664\,115} = 3$ years

Case study 2

Replace inefficient chilled water pumps with more efficient pumps in an office building.

Pump type : Centrifugal pump
Pump motor rating : 25 hp of 18.65 kW
Capacity : 951 gpm
Pump rated efficiency : 56%
Pump rated head : 80' of W
Operating pump suction pressure : 2 psi
Operating pump discharge pressure : 15.9 psi
Operating pump head : (15.9–2) × 2.307 = 32' of W
Operating efficiency of pump motor : 83%
Operating pump motor output : 14.65 kW

Operating water flow through pump : 1060 gpm
Operating pump efficiency : 43.7%
Total operating hours : 3000 hours (10 hours a day for 300 days)
Existing energy consumption of pump : 17.7 × 3000 = 53 100 kW
Rated efficiency of new pump : 0.7
Projected load on efficient pump for same head and flow : 10 kW
Projected energy consumption of new pump : 10 × 3000 = 30 000 kWh
Annual energy saving potential : 53 100 – 30 000 = 23 100 kWh
Annual saving in energy cost : 23 100 × 5 = Rs 11 5500
Initial investment required : Rs 45 000
Simple payback period : $\frac{45\,000}{115\,000} = 0.4$ years

Case study 3

Install variable speed drive on condenser pump to maintain constant entering condenser water temperature in a chilling plant in a luxury hotel in India.

Pump rating : 25 hp
Operating pump motor input : 17.7 kW

CDW (%)	*Hours (%)*	*Hours run*	*Operating kW*	*kW saved*	*kWh saved*
60	20	1486	3.82	13.9	20626
70	25	1858	6.07	11.6	21606
80	35	2601	9.06	8.6	22468
90	15	1115	12.90	4.8	5347
100	5	372	17.70	00.0	0

Total energy saving : 70 048 kWh
Total saving in energy cost : 70 048 × 5 = Rs 350 240
Initial investment required : Rs 420 000
Simple payback period : $\frac{\text{Rs } 420\,000}{\text{Rs } 350\,240} = 1.2$ years

Case study 4

Install wet bulb operated frequency drives at cooling tower fans in a luxury hotel.

Cooling tower motor rating : 12.5 hp
Operating kW of fan motor : 5.62 kW

Total operating hours in a year : 5675 hours
Operating airflow : 110 300 CFM
Required airflow : 80% of the operating flow
Modified load on fan : $(0.8)^3 \times 5.62 = 2.88$ kW
Saving in fan energy : $5.62 - 2.88 = 2.74$ kW
Annual energy saving potential : $2.74 \times 5675 = 15\,550$ kWh
Annual cost saving : $15\,550 \times 5 =$ Rs 77 750
Initial investment required : Rs 120 000
Simple payback period : $\frac{120\,000}{77\,750} = 1.5$ years

Case study 5
Install variable speed drives at kitchen exhaust and fresh air fans and schedule their operation in the main kitchen of a luxury hotel

Operating schedule	*Hz*	*CFM (%)*	*kW (%)*	*kW required*	*kWh consumed*
9 a.m. to 11 a.m.	35	70	40	14.8	10 804
11 a.m. to 3 p.m.	45	90	80	29.6	43 216
3 p.m. to 7 p.m.	40	85	65	24.1	35 186
7 p.m. to 1 a.m.	45	90	80	29.6	64 824

Exhaust fan rating : 30 hp
Fresh air fan rating : 25 hp
Operating load on exhaust fan/fresh air fan : 37 kW
Total energy consumed in existing ventilation systems : $37 \times 16 \times 365 = 216\,080$ kWh
Modified consumption after installation of VFD : 154 030 kWh
Total annual energy saving : $216\,080 - 154\,030 = 62\,050$ kWh
Total saving in energy cost : $62\,050 \times 5 =$ Rs 310 250
Initial investment required : Rs 250 000
Simple payback period : $\frac{250\,000}{310\,250} = 0.8$ years

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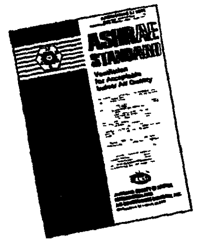

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IMPORTANT
Public interest in indoor air quality and health is growing fast. Health hazards posed by recirculating polluted air in air-conditioned spaces forms the basis for legal action in many building related law suits in the Western world.
And India may not be far behind.

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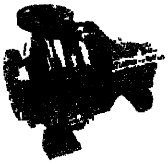

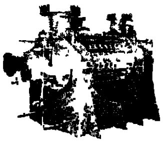

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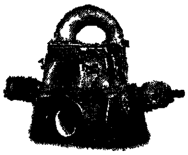

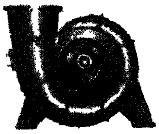

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Chapter 4

Pumps and fans

Managing energy efficiently in hotels and commercial buildings 91-103

Introduction

Pumps and fans are the mechanical means of transportation of fluid in the different systems and equipment in a hotel. Work has to be done by a prime mover in order to enable the pumps and fans to discharge its functions, because they are incapable of transporting the fluid on its own. The prime mover used in the hotel industry is mostly the electric motor.

Pumps

Pumps may be classified in several ways. Classification has been done on the basis of the general mechanical configuration. The detailed classification of pumps is given in Figure 1.

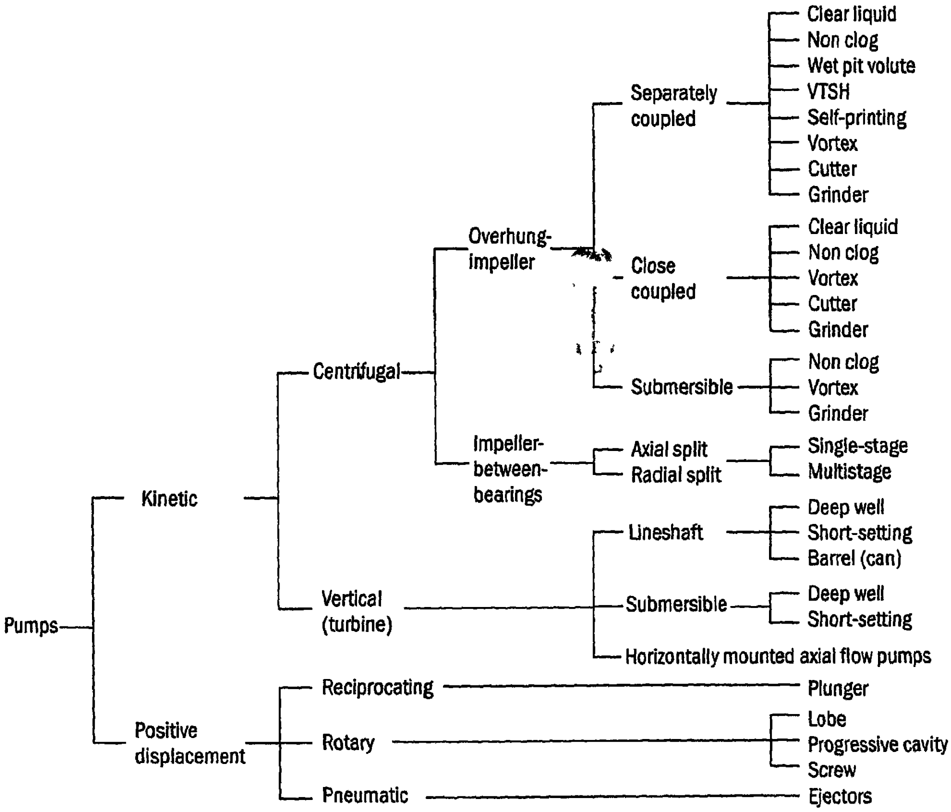

Figure 1 Classification of pumps

The two basic groups of pumps are (1) kinetic and (2) positive displacement. As kinetic pumps are used in most applications of the hotel industry, discussion in this chapter is confined to kinetic pumps.

Kinetic pumps

In kinetic pumps, velocity and pressure are imparted to the fluid as it moves past or through the pump impeller, converting some of the velocity into additional pressure. Kinetic pumps can be sub-divided into two major groups: (1) centrifugal (volute) pumps and (2) vertical (also called turbine) pumps.

Centrifugal pumps

All centrifugal pumps are equipped with a volute or casing. These pumps collect the liquid discharged by the impeller while converting some of the kinetic (velocity) energy into pressure energy. Centrifugal pumps based on mechanical configuration can be further sub-divided—overhung-impeller pumps and impeller-between-bearing pumps.

Overhung-impeller pumps

In an overhung-impeller pump, the impeller is mounted at the end of the pump shaft in a cantilever fashion. Both bearings are arranged on the same side of the impeller.

Impeller-between-bearing pumps

Impeller-between-bearing pumps fall in the group of axial split-case pumps, with the bearings mounted on each side of the impeller. Axial split-case pumps have a casing that is split along the (usually horizontal) centre-line of the shaft. The impellers can be readily exposed for inspection and service by removing the upper half of the casing.

Centrifugal pumps can be divided into three groups on the basis of the fluid movement through the pump into radial-flow pumps, mixed-flow pumps, and axial-flow or propeller pumps.

Vertical pumps

Vertical pumps are equipped with an axial diffuser (or discharge bowl) rather than a volute. The diffuser performs the same basic functions as the volute.

In vertical pumps, the bore size of the well limits the outside diameter of the pump and so controls the overall pump design. There are different types of vertical pumps—lineshaft pumps; submersible pumps; and horizontally mounted, axial flow pumps.

Lineshaft pumps
In this type of vertical pump, the lineshafting extends through the column to the bowl assembly and transmits torque to the pump rotor with the driver mounted on the discharge head.

Submersible pumps
In this type, the motor is mounted below the bowl assembly. The lineshaft is eliminated altogether and is replaced by a cable, which supplies power to the motor. Submersible pumps are useful for very deep (more than 180 metres or 600 feet) setting, for crooked wells, or where the noise of a motor would be objectionable.

Horizontally-mounted, axial-flow pumps
These are high capacity pumps and are typically used for flood control and similar applications. They are often engineered for each particular installation and may have many different driver and bearing arrangements.

Engineering parameters of pumping systems

A typical pumping system comprises a number of major equipment, which are shown in Figure 2. The pipe carries the liquid to be pumped from the reservoir to the site reservoir. Valves are incorporated to regulate or control the flow rate and pressure of the liquid being pumped.

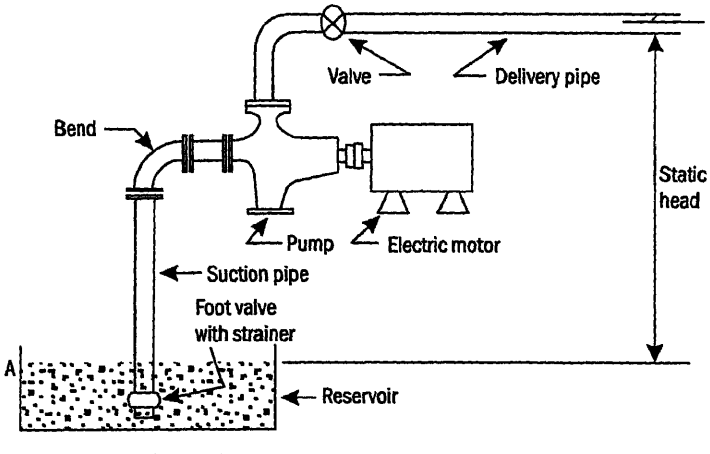

Figure 2 Typical pumping system

Head

The TDH (total dynamic head) is determined by adding the station suction and discharge head, the friction as head loss velocity head, and the fitting and valve head losses.

Any liquid that flows through pipes and other fittings like bends and valves experiences a resistance because of the frictional force opposing it. The equivalent length (m) for an additional pressure drop due to the valves and bends is given in Table 1.

Table 1 Equivalent length (in metres) for additional pressure drop due to valves and bends

Pipe diameter[a] (mm)	*Gate valve*	*Angle valve*	*Globe valve*	*Standard tee*	*Bend 90°*	*Bend 45°*
10	0.099	2.79	5.33	1.116	0.434	0.230
15	0.122	3.30	6.44	1.369	0.483	0.277
25	0.204	5.04	9.81	2.180	0.763	0.436
40	0.314	7.53	14.65	2.930	1.130	0.628
65	0.515	12.36	24.03	4.806	1.716	1.030
80	0.605	14.52	28.23	5.646	2.016	1.200
100	0.788	18.91	36.77	7.354	2.626	1.576
125	0.975	23.39	45.48	9.097	3.249	1.950
150	1.167	27.99	54.43	10.885	3.888	2.333

[a] nominal bore

Source Data obtained from Thermax Ltd

Capacity

Capacity is the volumetric quantity of liquid pumped per unit time. Usually, it is measured in cubic metres per hour or litres per second or gallons per minute.

Pump performance curve

The head produced by a pump at various flow rates and rotational speeds can be established in pump tests conducted by the pump manufacturer. During testing, the capacity of the pump is varied by throttling a valve in the discharge pipe, and the corresponding head is measured. The results of these tests and other tests with different impeller diameters are plotted to obtain a series of H–Q (head–discharge) curves for the pump at some given speed. Simultaneously, the power input to the pump is measured. The efficiency at various operating points is computed, and these values are also plotted in the same diagram. Together, these curves are known as pump characteristic curves.

System resistance curve

It is the plot of static head and friction head against Q. The static head is constant for all Q's and the friction head increases as Q increases. In Figure 3, 'SS' is shown as the system resistance curve. This curve is unique for a given pipe size, length and material and for a given number of bends and valves. Changes in any of the above parameters will make the curve shift. In Figure 3, it can be seen that 'SRC' (systems resistance curve) cuts the H–Q curve at point O. This point is called the operating point of the pump, i.e. wherever the SRC intersects the H–Q curve of a pump, the pump will operate at that point and deliver Q amount of liquid per unit time at head H. If the SRC shifts due to any of the reasons mentioned above, then the operating point will also shift to a point where the new SRC intersects with the H–Q curve.

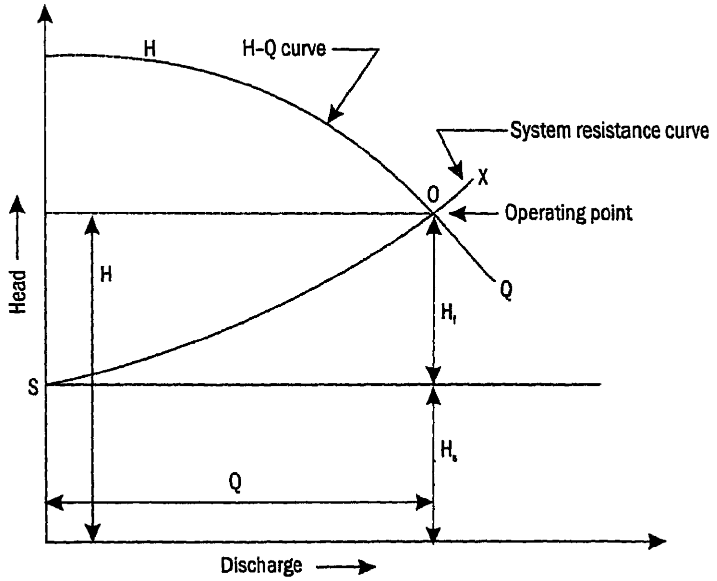

Figure 3 Pump operating point

Evaluation of pump performance

The efficiency of a pump is defined as the ratio of useful power output to the power input to the pump shaft.

The power output of a pump is the energy delivered to the fluid by the pump. It can be calculated from the formula

$$P_{out} = \rho QH$$

where

P_{out} = Power in watts

ρ = specific weight of fluid, N/m^3

Q = flow-rate, m^3/s

H = total dynamic head, m

The flow-rate can be obtained from the flow meters installed in the plant and the TDH determined by adding the station suction and discharge head, the friction as head losses, the velocity head, and the fitting and valve head losses.

$$TDH = H_{stat} + H_{ent} + H_{fs} + H_{fd} + \Sigma Hf_{vs} + \Sigma H_{fvd} + V^2_d/2g$$

where,

V_d - fluid velocity at the discharge end (m/sec).

H_{stat} - total static head, (m)

H_{ent} - entrance loss (m)

H_{fs}, H_{fd} - the frictional head losses, in suction and discharge piping system as computed from Hazen-Willians equation, (m)

ΣH_{fvs}, ΣH_{fvd} - headloss due to fittings and valves in suction and discharge piping system (m)

$V^2_d/2g$ - velocity head in the discharge pipe (m)

The actual power taken by the motor is measured with the help of a highly accurate digital power multi-meter and the power input to the pump shaft is obtained after considering the efficiency of the motor from its test report.

Pump efficiency

Pump efficiency is found by the following equation:

$$\text{Pump efficiency} = \frac{\text{Output}}{\text{Input}}$$

For an electric motor-driven pumping unit, the overall efficiency is found by the equation:

Overall efficiency = Pump efficiency × Motor efficiency

Operational problems in pumps

Cavitation

Cavitation does not occur during the normal operation of a pump. It appears only if the absolute pressure at any point in the pump, delivering a liquid liable to boil, is reduced to the vapour pressure of the liquid. It may cause permanent damage, reducing the performance of the pump. The pressure drop can be caused either by reduction in the suction head, increase in the suction lift or increase in the velocity of the liquid at certain points of its path through the pump. The effects of cavitation are two-fold: severe vibration, and erosion of the surface of materials used in the construction of the pump.

The effects of cavitational erosion are found not at the point of minimum pressure, but a little further upstream. It is, therefore, important to ensure that sufficient NPSH (net positive suction head) is available at the pump suction, and also that the liquid pumped does not contain an excess of free gas. For example, when a pump is operating on a water cooling duty, it is possible that although sufficient NPSH is available, the excess air absorbed by the water, when passing through the cooling tower, does not have sufficient time to be liberated. As the water passes through the pump, air is liberated in the low-pressure regions, causing cavitational erosion and vibration. NPSH is determined for handling water, but it is safer to use this figure without any correction for other liquids as well.

Option of improving pumping efficiency

Large safety margins are kept on the head of the pump, which results in the pump operating at a point that is not the specified duty point. This may cause increased power requirement in pumps using radial impellers. Figure 4 and Table 2 depict the initial duty point at A as specified by the buyer. Accordingly, a pump manufacturer supplied the pump. In reality, however, for a discharge requirement of Q, the pump was operating at point B (where SRC cuts the H–Q curve), whereas a pump with the duty point at C would serve the purpose of discharging required Q at required H.

Large differences are found in the efficiency of pumps manufactured by the organized sector and the small-scale sector. Selection should not be based on initial capital cost alone; efficiency and running costs should also be given due

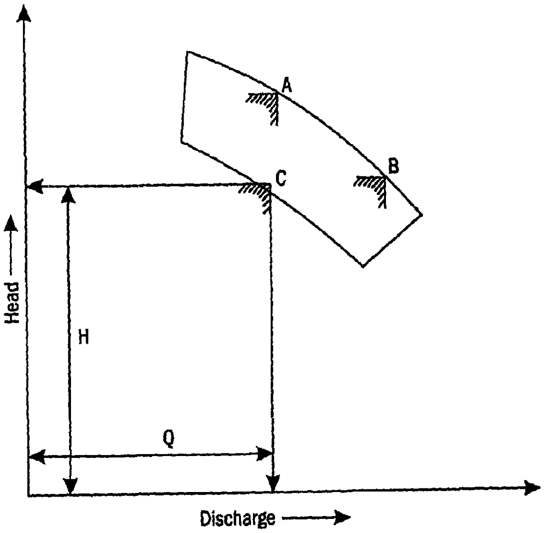

Figure 4 Pump operating point

Table 2 Operation parameter at different points of systems resistance curve

	Pump type		
Parameter	*A*	*B*	*C*
Head (m)	36	30	26
Discharge (litres per second)	75	90	75
Efficiency (%)	83	78	83

consideration. A proper system study should be done to match the system requirement with the highly efficient operation. In case the likely system's operating condition is unclear or is likely to be variable, effective methods of efficient modulation like variable speed drives should be introduced at the project stage itself. Alternatively, for fixed system requirements, options like change of pulley ratios or use of smaller impellers are possible.

Throttling

If the actual duty point does not coincide with the design duty point, then by means of throttling, i.e. partially closing delivery side valves, we can create an artificial head in the system and bring the pump to work near its design point. Its scope is limited and large variations are not possible. Therefore, every care should be taken to calculate the system head as accurately as possible. Despite this, if the actual head is different from design head, then throttling can be resorted to.

Changing impellers

For a model to cater to a certain well-defined range of head requirement, pumps are generally designed to accommodate different impeller sizes in the same casing. Throttling losses or problems can be avoided by replacing the existing impellers with new, smaller impellers.

Trimming impellers

Trimming of impellers can shift the pump characteristic downward to match the system head where the mismatch between the system and the pump is small. The flow and pressure will reduce, the efficiency also generally deteriorates marginally. Energy is saved as throttling losses or higher flow, as may have been the previous case, is avoided. The flow is directly proportional to the diameter of the impeller and the head is proportional to the square of the diameter of the impeller.

About 10% trimming may be possible in consultation with the manufacturer. The pump type, its performance characteristics, and the test certificate should be considered before trimming. The reason for this caution is that the manufacturer, in order to meet the purchase specification, may have already supplied the pump with a trimmed impeller or may have supplied it with the smallest impeller that is designed to operate in a particular casing. Further trimming may lead to a drastic drop in efficiency and may fail to meet the operating pressure and flow requirement.

Change of speed

The change of pump speed can be used to match the pump characteristics with the system characteristics. The head and power consumed are directly proportional to the square root and cube root of speed, respectively.

Pumps are usually directly coupled and speed variation implies change of electric motor, use of multi-speed motors or use of mechanical/electrical/electronic variable speed drives.

Replacement by appropriate pump

Replacement by a high efficiency pump at the required head of the system is desirable if the mismatch between the pumps and the systems is large. The payback periods are generally very attractive as only the pump is to be replaced, the existing electrical motor and switchgear can be retained as the change in motor efficiency due to change in shaft loading is likely to be marginal.

Fans

Fans are identified by two general groups: centrifugal, in which air flows radially through the impeller; and axial-flow, in which air flows axially through the impeller.

Centrifugal fans

There are three types of centrifugal fans, which are distinguished by the inclination of the fan blades. This inclination of the blades determines the performance characteristics of the fan. The three types are listed below.

1 **In forward-curved blade,** the tip of the blade is curved forward, i.e. in the direction of rotation of the fan wheel.
2 **The radial blade** has no curvature and the blade is straight and is fixed radially.
3 **In backward-curved blade,** the tip of the blade is curved backward or away from the direction of rotation of the fan wheel.

Centrifugal fans of the above three types are used for air conditioning and ventilation purposes.

Forward-curved fans

At any given rpm (revolutions per minute), the forward-curved blade fans deliver more air quantity (cubic feet per minute or cubic metre per hour) and produce relatively lower static pressure than the radial or backward-curved blade fans.

Radial blade fans

Radial blade fans are generally used in electric motors for internal cooling. The blades are built into the rotor itself in case of smaller-sized motors. For bigger-sized motors, a separate fan is fixed on the motor shaft (within the body).

Backward-curved fans

Backward-curved fans require higher speed and, therefore, have to be sturdier in construction. However, because of their characteristic higher efficiency over a broader range of system resistance, these types of fans are used for bigger installations.

Axial-flow fans

Axial-flow fans are of three types: (1) propeller, (2) tube axial, and (3) vane axial.

Propeller fans

Propeller fans are low-pressure high-capacity fans. The static pressure that these fans can develop is rather limited. They deliver large volumes of air with less horsepower than the centrifugal fans but can develop low static pressures. As the system resistance increases, the horsepower also increases. This is in contrast to the centrifugal fan in which the horsepower decreases with an increase in the system resistance. Thus a propeller fan used for an air-cooled condenser draws more electric current when the condenser fins get choked with dust. The air quantity delivered too goes down.

Tube axial fans

Tube axial fans may be considered as heavy-duty propeller fans mounted in a cylindrical housing. They are duct-mounted whereas the propeller fans are wall or diaphragm-mounted. Because of the high noise level produced by these fans, their application is limited to applications where the noise level is not of much consequence.

Vane axial fans

Vane axial fans are tube axial fans with vanes. The vanes located at the leaving side of the wheel help to straighten out the spiral flow of air coming out of an axial fan. These fans too are noisy.

The function of a fan is to increase the pressure of air which it handles by converting the relative velocity of air with respect to the blades at the entrance to pressure. The items that concern us in the study of fan performance are volume, pressure, and horsepower.

Fan performance

Manufacturers adopt a test set-up to establish the fan performance and to make fan-performance curves. The fan to be tested is connected to a long duct having a throttling device attached to its free end. By adjusting the opening of the throttling device, the air flow can be varied. The honeycombed air straightener is for straightening out the air flow in the duct. Manometer probes are connected to the duct after the straightener to obtain steady readings of the total and static pressures. A dynamometer is attached to the fan shaft to indicate the power consumed by the fan when it is in operation. When the fan is run with the throttling device fully opened, air flows in the duct without experiencing any resistance, i.e. no static pressure at all. So the total pressure reading on the manometer is just velocity pressure and air quantity handled by the fan is known as 'wide open cfm' or cmh – (WOcfm or WOcmh). If the speed of the fan is increased, the WOcfm will increase and consequently the velocity pressure will go up and vice versa. The air quantity is calculated by converting velocity pressure to velocity and then multiplying it with the cross-sectional area of the duct.

By adjusting the throttling device and varying the air outlet opening, the static pressure can be varied. The throttling device is progressively adjusted from the fully open to the fully closed position and readings of velocity pressure (total pressure – static pressure) and power consumed are taken and air quantities calculated for increasing steps of static pressures. By repeating this procedure for a number of fan speeds, air quantities and power consumed at various static pressures and fan speeds can be obtained. The dynamometer indicates the power required at the fan shaft and it is the BHP (brake horsepower) of the fan. The horsepower of the fan motor has to be higher than the BHP to take care of the drive-losses in motor efficiency, and to overcome the initial starting inertia.

When the throttling device is fully closed, the air flow in the duct ceases completely, and then no velocity pressure will be indicated. Only static pressure will be shown and this is known as the BTSP (Block Tight Static Pressure).

Usually fan performance graphs/charts include curves intersecting the air quantity lines to show the air quantities as a percentage of the WOcfm/WOcmh. Also, a surge line is drawn (Figure 5). Operation of the fan at points to the left of the surge line will be unstable with highly fluctuating air delivery and static pressure, resulting in noise and vibration.

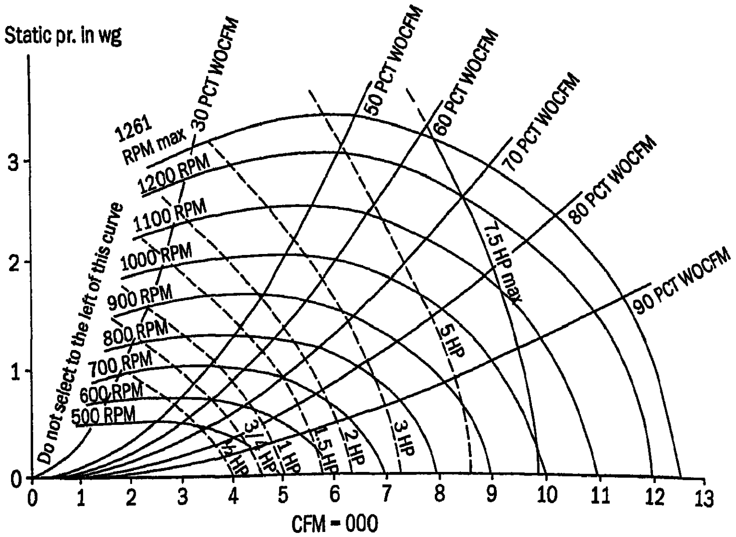

Figure 5 Fan performance curves

With the help of the fan-performance curves, the behaviour of the fan in the air distribution system for which it is selected can be predicted. This is done by superimposing the system resistance curves on the fan curves.

Improving performance

Optimizing size of the fan

In many cases, fans are selected anticipating extremely large system resistance. However, this may not be the case in actual practice. Often over-sizing of fans results in operating inefficiencies. In such cases, operating efficiency can be improved by replacing the existing fan with a lower capacity fan.

Change of fan speed

The change of fan speed can be used to match fan characteristics with system characteristics. The head and power consumed are directly proportional to the square root and cube root of speed, respectively.

Fans are usually directly coupled and speed variation implies change of electric motor, use of multi-speed motors or use of mechanical/electrical/electronic variable speed drives.

Parameters and variables relating to energy performance of fans

Static efficiency

Static efficiency denotes the efficiency of a fan and is defined as the ratio of static energy obtained to the energy input and is expressed as a percentage.

$$\text{Static efficiency \%} = \frac{\text{Power output}}{\text{Power input}} \times 100$$

or

$$\frac{\text{m}^3\text{/min} \times \text{SP (mm)}}{4843 \times \text{BHP}}$$

or

$$\frac{\text{cfm} \times \text{static pressure (inch)}}{6356 \times \text{BHP}}$$

For a given flow-rate of air, a slightly bigger fan is more efficient than a smaller fan, both handling the same air quantity and static pressure. The bigger fan will run at a lower speed and so the noise level, too, will be less.

Air horsepower (Ahp)

Air horsepower is the power required to move a volume rate of air through a head (total pressure).

$$\text{Ahp} = K \times Q \times P_1$$

where Q = air flow rate in m^3/min or cfm

P_1 = Total pressure in mm WG (water guage) or inches WG

And K = 0.000218 or 0.000157

Mechanical efficiency of the fan = Ahp × 100/BHP

Fan laws

Speed

The quantity of air (Q) delivered by a fan, the static pressure against which the fan can deliver this quantity of air, and the horsepower required are dependent upon the speed of the fan.

- The air quantity (Q) delivered by the fan varies as the fan speed (rpm).
- The static pressure developed by the fan varies as the square of the speed.
- The horsepower required varies as the cube of the speed.

These can be expressed in the form of equations as follows. Q_2 and Q_1 are air quantities; SP_2 and SP_1 are the static pressures; hp_2 and hp_1 are horsepowers of the fan at two different speeds N_2 and N_1.

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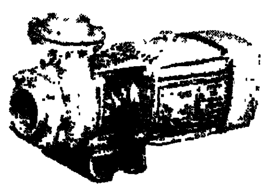

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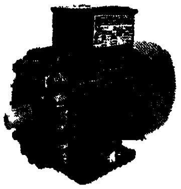

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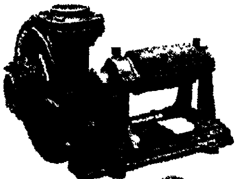
MOTOR



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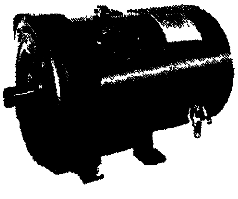

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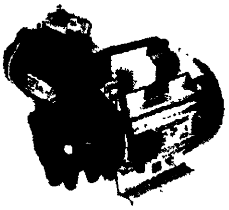

KIRLOSKAR **STAR** PUMP

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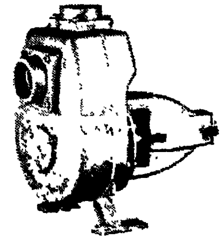

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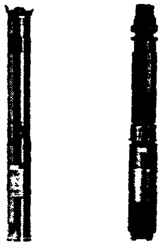
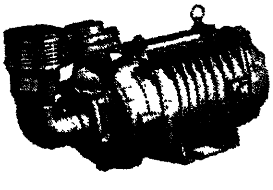


100mm(4") 150mm(6")

BOREWELL SUBMERSIBLE PUMP



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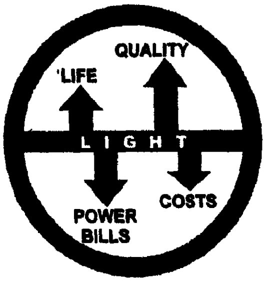

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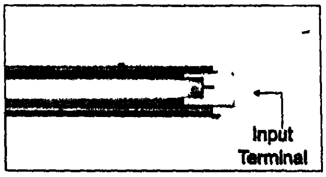
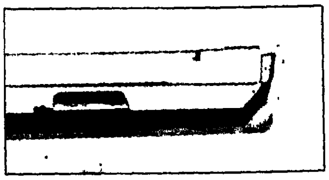


Stand alone



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Chapter 5

Lighting systems

Managing energy efficiently in hotels and commercial buildings 107-121

In commercial buildings, lighting is a major component of the building energy cost. In a fully air-conditioned office, lighting accounts for about 30% of the electricity used, and in non-air-conditioned offices, it is even more than the energy used in space conditioning. Hospitals consume 20%–30% of the electricity for lighting. In luxury hotels, lighting accounts for 25%–40% of the total energy use. The connected load in commercial building varies from $15\ \mathrm{W/m^2}$ to $25\ \mathrm{W/m^2}$.

Assessment of lighting energy efficiency and energy consumption

The energy consumed by a lighting installation depends upon the power consumption of the luminaires and the length of time for which they are switched on. Both of these aspects are important, as changes in either will affect the energy efficiency of the installation. It is important to know the energy consumption of an existing or proposed lighting installation when considering the cost-effectiveness of measures to improve its energy efficiency.

Installed load

The installed lighting load should be estimated by counting the number of each type of luminaires and then multiplying it by the power of the luminiares. Luminaire's power is the sum of lamp and ballast power. For new installations, the manufacturers' data should be used. In older installations, for estimation of load in fluorescent fixtures with conventional wire-wound ballasts, the total lamp power should be multiplied by 1.1; for high intensity discharge lamps of less than 250 W, it should be multiplied by 1.25. For filament lamps (incandescent, halogens), the rated lamp power alone should be used. For fluorescent lamps controlled by high frequency electronic ballasts, the manufacturers' data should be used, but generally the circuit watts will be equal to the watts worked on the lamps.

The total installed load estimated should be divided by the floor area of the building to determine the loading in watts per square metre. This can be compared with typical valves of good lighting practice recommended by various lighting codes to measure the efficiency of the lighting schemes. A lighting scheme with loading in the range of 8 to 12 W/m^2 can be considered efficient.

Length of usage

The usage of a lighting installation depends upon the occupancy patterns of the space, the daylight available in the space, and the control system used. In existing installations, the estimation of actual usage can be made by using a simple hoax-run meter connected to the lighting circuits or by data loggers. Estimates of the total lighting use can often be made from the results of monitoring a representative sample of the lighting in a building. In some buildings, simple observations of hours of use can be made by a person visiting the space in a planned way over a period of time.

The energy consumption can then be derived from the total lighting load multiplied by the hours of use of the installation. For lighting installation that can be dimmed, it is not possible to derive energy consumption simply from total installed load and hours of use, since the energy consumption at any time will depend on the actual level of output of the lamps at the time. It may be possible to estimate a maximum likely consumption, but reliable data can only be derived by monitoring the electricity consumed.

Lighting installations

The basic requirement of any lighting installation in a commercial building is to provide right light at the right place at right time. The quantity and quality of light determines the 'right light'.

Light quantity - illuminance

A common misperception contributing to the proliferation of ineffective and inefficient lighting is that more light equals higher quality light. Lighting level requirements have evolved with changes in workplace and knowledge of visual science. The BIS (Bureau of Indian Standards) and the IESNA (Illuminating Engineering Society of North America) have developed lighting standards to select appropriate illuminance levels for various indoor activities in different commercial buildings. For any area in commercial buildings, lighting levels should conform to these standards (Table 1).

Light quality

A right light should have right quality of light. Sometimes it happens that though the quantity of light in a lighting scheme conforms to the recommended standard, it fails to provide the required visual comfort since it does not conform to the recommended light quality. Glare, colour rendering, and colour appearance are main criteria, which decide the quality of light in a commercial building.

Table 1 Right lighting levels for different tasks

Need/Task	*Location*	*Levels (lux)*
Safety/security in an unoccupied outdoor area	Public spaces with dark surroundings	30
Visibility for short visits or walk-through	Corridors, inactive storage areas, dining areas in restaurants	75
Service-oriented work with minimal visual tasks	Office and hotel lobbies, elevators, stairways, washrooms, active storage	150
Shop or office work that involves high-contrast or large-size visual tasks	Conference rooms, general office and shop areas, cashiers	300
Shop or office work that involves medium-contrast or small-size visual tasks	Office desk areas, machine shop task lighting, restaurant kitchens	500-750
Drafting, general health care examinations, other tasks of low contrast	Operating rooms and health care facilities, drafting rooms	750-1500

Glare

Glare creates discomfort. Glare results when luminance levels or the differences in luminance levels are too high, and objects appear too bright. As glare leads to loss of visual performance and impaired visibility, it should be minimized whenever possible.

Generally two types of glare are encountered – direct glare and reflected glare. Direct glare occurs when light from a bright object enters the eye directly. It can be controlled through the use of luminaire lenses, louvers, and window blinds – all of which block the direct viewing of sources. Reflected glare occurs when reflected light creates a shining or veiling reflection, which reduces or eliminates task contrast. This commonly occurs on shining, light-coloured surfaces and computer screens. Although veiling reflections are most difficult to control, they can be minimized by moving the light source, reorienting the task, and installing reflectors, lenses or louvers or luminaires. Good general practices to minimize glare include the use of lower ambient light levels, task lighting, indirect lighting, and luminaires with a high VCP (visual comfort probability) rating. The VCP index provides an indication of the percentage of people in a given space that would find the glare from a fixture to be acceptable. A minimum VCP rating of 70 for commercial interiors and 80 for computer areas should be ensured.

Colour rendering

The effect a light source has on the appearance of CRI (colour rendering index) ranges between 0 and 100. A perfect CRI rating (100) means that a lamp's light is identical to daylight in colour quality. Lamps with a higher CRI make people and objects appear more natural and bright, and hence improves visual clarity and aesthetics.

Colour temperature

The colour temperature ranges from warm to cool and is a common method to describe the kind of light thrown. A candle throws very warm light while daylight fluorescent is at the cool end of the spectrum. This spectrum is measured in Kelvin (K) with higher temperatures indicating cooler light sources. Most lighting sources range from 1800 K (a warm candle) to 4000 K (a cool white fluorescent lamp).

Lighting equipment

Lamps

Incandescent lamps

Incandescent lamps are the oldest electrical light source. In these lamps, light is produced by the electric heating of a filament to such a high temperature that radiation in the visible region of the spectrum is emitted. GLS (general lighting service) lamps form the mainstay of incandescent lamps. The bulb is either clear, fronted or opal and wattage ranges from 15 W to 200 W, with the great majority lying between 25 W and 200 W. GLS lamps have shorter life span (between 750 hours to 1000 hours) and are not very efficient. The luminous efficacy (defined as light output [lumens] divided by wattage and considered as a measure of lamp efficacy) varies between 8 and 21.5 lm/W. Light emitted by these lamps is warm (colour temperature is around 2800 K) and its colour-rendering ability, with its CRI as 100, is second to none (Table 2).

Tungsten halogen lamps

The basic difference between a GLS and a tungsten halogen lamp is that of a halogen which is added to the normal gas filling and work on the principle of a halogen regenerative cycle to prevent blackening. Tungsten halogen lamps are characterized by high bulb temperature, smaller size, and luminous efficacy around 10% more than the normal incandescent lamps. Tungsten halogen lamps for normal lighting purposes have a colour temperature of between 2800 K and 3200 K, and with its CRI of 100 provide excellent colour rendering (Table 2).

Fluorescent lamps

The fluorescent lamps are low-pressure mercury discharge lamps in which light is produced predominantly by fluorescent powders activated by the ultra-violet rays of the discharge. The lamp is in the form of a long tubular bulb with an electrode sealed into each end, contains mercury vapour at low pressure with a small amount of inert gas for starting and arc regulation. The inner surface of the bulb is coated with a fluorescent powder or phosphor, the composition of which determines the quantity and colour of the light emitted. Fluorescent lamps in India are available in four forms: (1) standard fluorescent lamps, (2) slim fluorescent lamps, (3) higher efficiency slim fluorescent lamps, and (4) T-5 lamps.

- *Standard fluorescent lamps (T-12)* Standard fluorescent lamps are 40 W lamps with a diameter of 38 mm. Their lumen output varies between 2450 and 2770 and colour temperature from warm water (4300 K) to cool daylight (6500). 20 W and 80 W lamps are also available (Table 2).
- *Slim fluorescent lamps (T-8)* The diameter of a slim fluorescent lamp is 26 mm and the wattage is 36 W. Lumen output is 2450 and colour temperature 6500 K (cool daylight). They are also available for 18 W (Table 2).
- *Higher efficiency slim fluorescent lamps (Trulite)* Higher efficiency slim fluorescent lamps are of 18 W and 36 W and manufactured in three different colours – warm (2700 K), bright white (4000), and cool daylight (6500 K). They are based on the highly sophisticated tri-phosphor fluorescent powder technology and offer excellent colour-rendering properties (CRI = 85). Their high luminous efficacy (more than 30% of the standard 40 W fluorescent lamp) saves initial system cost, annual operating cost, and reduces air-conditioning load while the high lumen output leads to better lighting condition (Table 2).
- *T-5 lamps* T-5 lamps are fluorescent lamps with a diameter of 16 mm, which is 40% thinner than the diameter of existing slim fluorescent lamps. They are designed for higher efficacy and system miniaturization. The tri-phosphor layer in combination with precoating and lamp load add up to a high efficacy (lumen output of 28 W lamp is around 2900 lumens). They are available in 14 W and 28 W and have colour range from incandescent light (2700 K) to cool daylight (6500 K). The lamp life of T-5 lamp is also long – around 10 000 hours as compared to 8000 hours of standard fluorescent lamps (Table 2).

Compact fluorescent lamps

The CFLs (compact fluorescent lamps) combine fluorescent efficiency with incandescent convenience. They have high efficacy, good colour characteristics, low energy consumption, and long life (typically 8000 hours as against the 1000 hours of a GLS lamp), which make them an ideal replacement for GLS lamps (Table 2).

High-intensity discharge lamps

The HID (high-intensity discharge) lamps provide the highest efficacy and longest service life of any lighting type. They are commonly used for outdoor lighting and in large indoor areas. HID lamps use an electric arc to produce intense light. They also require ballasts and take a few seconds to produce light when first turned on because the ballast needs time to establish the electric arc.

The three most common types of HID lamps are mercury vapour, metal halide, and high-pressure sodium. HID lamps and fixtures can save 75% to 90% of lighting energy when they replace incandescent lamps and fixtures. Significant energy savings are also possible by replacing old mercury vapour lamps with newer metal halide or high-pressure sodium lamps (Table 2).

Mercury vapour lamps

The mercury vapour lamp is the oldest type of HID lighting and is used primarily for streetlighting. Mercury vapour lamps provide about 50 lumens per watt. They cast a very cool blue/green-white light. Most indoor mercury vapour lighting in sports arenas and gymnasiums has been replaced by metal halide lighting, which has better colour rendering and efficiency (Table 2).

Metal halide lamps

Metal halide lamps are similar in construction and appearance to mercury vapour lamps. The addition of metal halide gases to mercury gas within the lamp results in higher light output, more lumens per watt, and better colour rendition than from mercury gas alone. Metal halide lamps are used to light large indoor areas such as gymnasiums and sports arenas, and for outdoor areas such as car parks or any other places where the colour rendition is important (Table 2).

High-pressure sodium lamps

The high-pressure sodium lamp is fast becoming the most common type of outdoor lighting. It provides 90 to 150 lumens per watt and has efficiency exceeded only by low-pressure sodium lighting. These lamps are reliable and have a long life. Their colour is a warm white, and their colour rendition ranges from poor to fairly good, depending on design and intended use (Table 2).

Low-pressure sodium lamps

Low-pressure sodium lamps work in some ways like fluorescent lamps. They are the most efficient artificial lighting, have the longest service life, and maintain their light output better

Table 2 Characteristics of light sources

Lamp type	*Luminous efficacy* (lm/W)	*Lamp life* (hours)	*Colour temperature* (°K)	*Colour rendition* index
GLS	11-19 (Typical: 12)	1000	2700	100
Tungsten halogens				
Line voltage	11-17 (single ended)	2000-3000	3000	100
	14-23 (double ended)	2000-3000	3000	100
Very low voltage	12-22 (Typical: 18)	2000-4000	3000	100
Fluorescent				
T-12	45-100 (Typical: 70 with copper choke)	8000	3000-4100	60-85
T-8	100 (Triphosphor on electronic ballast) 97 (Triphosphor on copper choke) 77 (Halophosphate on copper choke)	8000	2700-6500	50-98
T-5	80-100	8000	3000-6000	85
CFLs				
Integral (self ballasted)	30-65	10000	2700-4000	85
Dedicated (pin-based)	85-98	10000	2700-6000	85-98
Metal halide	65-120 (Typical: 70)	3000-20000	2900-6000	60-93
High pressure mercury	32-60	10000-24000	3900-4300	33-50
High pressure sodium				
Standard	65-150 (Typical: 110)	10000-24000	2200	20-65
White sodium	57-76 (Typical: 65)	15000	2500	80
Low pressure sodium	100-200	12000-24000	1700	0

than any other lamp type. Low-pressure sodium lighting is used where colour is not important because it renders all colours as tones of yellow or grey. Its typical applications include are in highways and for security lighting (Table 2).

Ballasts The fluorescent lamp has a negative resistance characteristics and, therefore, it must be operated in conjunction with a current limiting device (or ballast) to prevent current runaway. The commonly used ballasts, which have positive resistance characteristics, are a choke or inductor and an electronic circuit.

Inductive ballasts

An inductor or choke is the most widely used ballasts for normal AC applications. It consists of a large number of windings of copper wire on a laminated iron core. Heat losses, which occur through the ohmic resistance of the windings, depend on the mechanical construction of the ballast and the diameter of the copper wire. Approximately 7 W to 15 W power loss takes place in the choke.

Electronic ballasts

Electronic ballasts are more expensive as compared to copper chokes but they offer important advantages over conventional chokes – improved lamp and system efficiency, no flickering, instant starting without a separate starter, increased lamp life, excellent light-regulation possibilities, no need for power factor correction, etc. The power loss in electronic ballasts is approximately 1 to 2 W and they are an excellent replacement to copper chokes in fluorescent lamp fixtures.

uminaires Luminaires or light fixtures are used to direct light efficiently while providing a high level of visual comfort. Getting a large percentage of light to exit the fixture while controlling its distribution usually requires a compromise. Generally, the most efficient fixtures have the poorest visual comfort. Conversely, fixtures with excellent glare control are the least efficient. In new light schemes, the best compromise between the fixture efficiency and visual performance should be first determined and based on that the best fixtures should be selected, which also fits into the architectural design objectives. But when retrofitting, fixture reflectors and shielding materials should be added to achieve these objectives. Reflectors in luminaires are used to reduce the internal light loss in fixtures by using highly reflective surfaces to redirect light out of the fixture. In retrofits, reflectors improve fixture efficiency by improving the internal surface reflectance up to 17%.

Most indoor commercial fixtures use some type of diffusers, lens, or louver over the face of the fixture to block direct view of the lamp or to diffuse or redirect light. Diffusers are semi-translucent plastic sheets that hide lamp and diffuse light evenly across the face of the fixture. Because diffusers absorb light and distribute light in all directions, they are very inefficient and also ineffective at controlling glare. By using clear plastic lenses with small prismatic surface patterns, both fixture efficiency and light distribution can be improved better as compared to diffusers.

Louvers depending on cell size and depth can provide a better balance between superior light control and energy efficiency. Small paracube louvers (cells less than 2.5 cm), although they provide excellent glare control, are quite inefficient and they should be avoided. Layer deep cell louvers provide high efficiency and excellent light control. New parabolic louvered fixtures are designed to combine high efficiency (90%) with very high VCP ratings above 90 and by far they are the most efficient fixtures for indoor lighting (Table 3).

Table 3 Luminaire efficiency and glare control

Shielding material	*Luminaire efficiency range* (%)	*VCP range* (%)
Stand clear lens	60-80	50-70
Low-glare clear lens	60-80	75-85
Deep cell parabolic louver	50-90	75-99
Translucent diffuser	40-60	40-50
White metal louver	35-45	65-85
Small cell parabolic louver	40-65	99

Lighting controls

Reducing the connected load of the lighting system represents only half of the potential for maximizing energy savings. The other half is minimizing the use of that load through appropriate lighting controls. There are a wide range of choices available today from simple light switches to fully automated systems. Automatic controls switch or dim lighting based on time, occupancy lighting-level strategies, or a combination of all three.

Timers

Timers are the most basic type of automation, and are very popular for outdoor applications. Timers can be simple (responding to one setting all year round), or sophisticated enough to contain several settings that go into effect over time.

Occupancy sensors

Occupancy sensor devices, also known as 'motion detectors', turn lights off and on in response to human presence. Occupancy sensors can be calibrated for sensitivity, and have multiple settings that govern when lights are extinguished. Once sensitivity and coverage area are established, sensors are selected from two predominant technology types.

Passive infrared sensors detect the motion of heat between vertical and horizontal fem patter detection zones. This technology requires a direct line of sight and is more sensitive to lateral motion, but it requires layer motion as distance from the sensor increases. The coverage pattern and field of view can also be precisely controlled. It typically finds its best application in smaller spaces with a direct line of sight.

Ultrasonic sensors detect movement by sensing disturbances in high-frequency ultrasonic patterns. Because this technology emits ultrasonic waves that are reflected around the room surfaces, it does not require a direct line of sight, is more sensitive to motion towards and away from the sensor, and its sensitivity decreases relative to its distances from the sensor. It also does not have a definable coverage pattern or field of view. These characteristics make it suitable for use in layer enclosed areas that may have cabinets, shelving, partitions, or other obstructions. If necessary, these technologies can also be combined into one product to improve detection and reduce the likelihood of false on or off triggering.

Photocells

Photocells measure the amount of natural light available and suitable for both indoor and outdoor applications. When available light falls below a specified level, a control unit switches the lights on (or adjust a driver to provide more light). Photocells can be programmed so that lights do not flip on and off on partially cloudy days.

Energy management systems

Also known as building automation systems, these computer-driven systems along with controlling all building loads also control lighting. Energy management systems are most effectively employed in the initial construction of a building, not as a retrofit application. By using a variety of control units tied to a centralized system, the building monitors its lighting system itself. However, settings can usually be overridden manually.

Maintenance

Over time, luminaires and room surfaces get dirty and the light output from lamps decreases. The light output from a lighting installation also decreases. Lack of maintenance leads to poor performance of a lighting installation. Simple cleaning of lamps and luminaires can often substantially improve the lighting of the space and help reduce energy wastage. Maintenance requirements should be considered while designing installations. Luminaires that are easily accessible are more likely to be cleaned regularly and have their lamps replaced. Proper cleaning materials and techniques should be used to minimize losses caused by chemical action or scratching of optics and electrostatic dust accumulation.

Potential energy-saving actions

No-cost measures

Remove unnecessary lamps

Lamps in places where lighting is too bright, especially near windows, in hall ways, and in areas with no furniture should be removed. Lowest lighting levels should be determined based on the recommended lighting levels as suggested by recognized lighting codes with the aid of a light or lux meter.

In multiple-lamp fluorescent fixtures, lamps are wired in pairs and, therefore, must be removed in pairs.

Disconnect ballast

If the lamps are removed but their ballast is not disconnected, the fixture will continue to use some electricity when it is switched on. For maximum energy saving in fixture after delamping, the ballast should also be disconnected.

Turn off lights near windows

If lights can be controlled separately, turn off those nearest the windows whenever there is enough natural light. If present wiring and switches do not allow this, it is better to rewire.

Turn off lights when an area is unoccupied

Lights should be turned off whenever an area is left unoccupied for any length of time. Consider using a label or sign next to the light switch to remind people to turn it off. Although the life of a lamp is shortened if it is turned on and off frequently, it is to be remembered that the cost of the lamp is insignificant when compared to the cost of energy the lamp will consume during its lifetime.

Re-schedule evening cleaning

Cleaning hours are generally after evening hours, when lights are on only for this purpose. Sometimes, because of the wiring and switches the entire floors and even the whole building must be lighted even though only one area is being cleaned at a time. To the extent possible, re-schedule evening cleaning to times when the lights are on anyway because the area is in use, or during day time.

Keep lamps and fixtures clean

Dust, grease, and other dirt accumulations can absorb as much as 30% of the light from lamps and reflecting surfaces. Clean them on a regular basis.

Use or eliminate unnecessary outside lighting

Buildings have lighted painting areas, signs, entrances, façade, walls, and landscaping. It may be possible to eliminate some of these lighting entirely or use of lower wattage or more efficient lamps or reduce the length of their usage.

Use only necessary safety and security lighting

Unnecessary lights in pathways and other areas can waste considerable energy. Use lower wattage bulbs if appropriate. At night and when areas are unoccupied, make sure the only lights kept on are those needed for safety, security, or some other specific purpose.

Possible investment measures

Install high efficiency fluorescent lamps to replace standard ones

The lumen output of 40 W fluorescent lamp is around 2450. The 36 W slim tube also has the same lumen output but it saves 4 W per lamp. The 36 W Trulite with triphosphorus coating has 3250-lumen output, which would give 30% extra light. Now 28 W extra slim T-5 tubelights are available in the market with light output of 2900 lumen and longer lamp life. Replacement of 40 W fluorescent lamp with T-5 fluorescent lamp saves around 25 W per lamp.

Replace conventional ballast with conventional ballast

Power loss in copper wound chokes in fluorescent lamp is 15 W per ballast, and in high frequency ballast it reduces to 1–3 W. Replacement of copper choke in a fluorescent lamp fixture can save up to 25% energy being consumed in the fixture.

Replace inefficient incandescent lamp with compact fluorescent lamps

The lumen efficacy of GLS is very low and their replacement with appropriately sized CFLs, which is one-fifth of the wattage of GLS, saves around 80% energy in lighting installation.

Install dimmer switches

Different amounts of light may be needed at different times or in different parts of an area. The simplest way of providing this flexibility may be to install one or more dimmer switches. Energy-efficient solid-state dimmers are recommended, not the old-fashioned wasteful rheostat type.

Install timers for external lighting

Installation of time clocks in external lighting will switch the light on and off automatically at desired timings. Timers that are activated by photocells will automatically adjust for seasonal variations in daylight. Timers that are set by hand to operate at fixed predetermined times must be readjusted seasonally.

Install sodium lights for parking, other areas

Sodium lights make colours appear quite different, since colour-rendering power of these is extremely poor. However, the very high energy efficiency of these lamps makes them an excellent choice for parking areas and for other exterior lighting.

Table 4 Recommended energy conservation measures for buildings

Energy conservation measures	*Offices*	*Education*	*Hospital*	*Retail*	*Hotel*
Optimize day light	✓	✓	✓	✓	✓
Replace GLS with CFLs	✓	✓	✓	✓	✓
Replace GLS display with tungsten halogen or high pressure discharge	✓	✓	✓	✓	✓
Replace 40 W TL with 28 W fluorescent lamps	✓	✓	✓	✓	✓
Install more efficient luminaires	✓	✓	✓	✓	✓
Install high frequency ballasts	✓	✓	✓	✓	✓
Regularly maintain and clear luminaires and lamps	✓	✓	✓	✓	✓
Localized lighting instead of general lighting	–	–	✓	✓	✓
Encourage manual switch-off when lighting is not needed	✓	✓	✓	✓	✓
Fit localized switching close to task area	✓	✓	✓	✓	✓
Install automatic lighting controls	✓	✓	✓	✓	✓

Table 5 Cost-benefit analysis of energy conservation options

Energy-efficiency measure	*Cost* (Rupees)	*Energy savings* (%)	*Payback period* (years)
Replace 38 mm tubes (40 WTL) with 26 mm tubes (28 W-T05)	800/lamp	45%	<2
High frequency ballasts for fluorescent lamps	400/lamp	25%	<2
Replace GLS with CFLs	650/lamp	75%	1-2
Replace opal or prismatic diffusers with specular reflectors	3000/fixture	If fewer luminaires are needed	
Install automatic lighting controls	–	2-50	2-5
Localized lighting to supplement general lighting	–	60-80	4-8

Case studies

Case study 1

Energy and cost-saving potential due to replacement of 40 W fluorescent lamps (with ordinary choke) with 36 W fluorescent lamp (with electronic choke) in an office building.

Number of fixtures (twin) : 500
Number of lamps : 1000
Existing connected load : (50 + 15) × 1000/1000 = 65 kW
Modified connected load : (36 + 2) × 1000/1000 = 38 kW
Saving in load : 55 – 38 = 17 kW
Total operating hours/year (10 hours/day for 300 days in a year) : 10 × 300 = 3000
Total energy saving potential : 17 × 3000 = 51 000 kWh
Total saving in energy cost (Electricity tariff Rs 5/kWh) : 5 × 51 000 = Rs 255 000
Investment required (Rs 40 for lamp and 400 for ballast) : 440 × 1000 = 440 000
Simple payback period : 440 000/255 000 = 1.62 years

Case study 2

Replacement of 36 W fluorescent lamps with copper choke with 28 W fluorescent lamp in a luxury hotel kitchen.

Number of fixtures (twin) : 460
Number of lamps : 460 × 2 = 920
Hours of use : 24 hours for 365 days
Existing lighting load : (36 + 15) × 920/1000 = 46.92 kW
Modified lighting load : (28 + 2) × 920/1000 = 27.6 kW
Reduction in lighting load : 46.92 – 27.6 = 19.32 kW

Annual energy saving potential : 19.32 × 24 × 365 = 169 243 kWh
Saving in energy cost : 169 243 × 5 = Rs 846 216
Initial investment required (cost of lamp @ Rs 800/lamp) : 800 × 920 = Rs 736 000
Simple payback period : 736 000/846 216 = 0.9 years

Case study 3
Replacement of GLS with CFLs in a luxury hotel lobby.

Number of fixtures : 86
Number of lamps (two lamps in each fixture) : 86 × 2 = 172
Wattage of installed lamp : 60 W
Existing lighting load : 60 × 172/1000 = 10.32 kW
Recommended CFL wattage : 15 W
Modified lighting load : 15 × 172/1000 = 2.58 kW
Reduction in lighting load : 10.32 – 2.58 = 7.74 kW
Operating pattern of fixtures (average 16 hours/day for 365 days) : 16 × 365 = 5840 hours
Annual energy-saving potential : 7.74 × 5840 = 45 201 kWh
Annual saving in energy cost : 45 201 × 5 = Rs 226 005
Initial investment required (lamp cost @ Rs 650/lamp) : 650 × 172 = Rs 111 800
Simple payback period : 111 800/226 005 = 0.5 years

Case study 4
Installation of occupancy-linked sensors in lobby toilet in a luxury hotel

Total lighting load in lobby toilet (ladies/gents) : 2.3 kW
Operating hours/day : 14
Operating hours/day after installation of sensor : 6
Total energy saving potential : 6 × 2.3 × 365 = 5037 kWh
Annual saving in energy cost : 5037 × 5 = Rs 25 185
Initial investment required (Rs 8000/sensor) : Rs 16 000
Simple payback period : 0.7 years

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- Energy Efficiency Study
- M & V Protocol
- Renewable Energy
- Cogeneration
- Clean Technology
- EMS
- QMS
- TQM

Chapter 6 Cogeneration systems

Managing energy efficiently in hotels and commercial buildings 123-129

Introduction

Considering the present-day availability and reliability of electric power supply from the grid and its cost, there is a pressing need for industry for energy conservation and efficient use of energy more than ever before. Electric power production through large power plants and distribution to users spread out far and near is not geared to the exigencies o energy shortages and energy conservation. A good percentage of energy is first rejected through cooling towers during generation and then through T&D (transmission and distribution) network, before the power reaches the end-user. Thus, a great deal of attention has been paid by industry to adopt a variety of heat recovery systems operating under the principle of total energy concept. The total energy system or cogeneration implies on-site power generation using steam turbines, gas turbines/engines or diesel generator and taking the steam requirements for the process from the exhaust of the back-pressure steam turbine or from the heat recovery steam generator. In some cases, the power generated may be in excess of the in-house requirements by the industry. In that case, the excess power can be exported to the local power grid through a suitable agreement with the electricity distribution company.

Cogeneration application for hotel industry/ commercial buildings

Typical cogeneration plants

The sizing or size selection of a cogeneration system is determined by the economic attractiveness of the system in a given industrial process application. First of all, the fuel to be used (natural gas or oil) is to be defined. Then, the electric load in terms of kilowatts and thermal load in terms of kilocalories (for heating and chilling applications) are defined. For simplicity, the output of electricity in kilowatts is used to indicate the size of any given system.

The fuel available for hotels or commercial buildings is normally diesel or natural gas or both. The electric power requirement is up to 3500–4000 kW in addition to thermal load in the form of steam and hot water, and the chilling load for conditioning the air. These loads can be met through a typical cogeneration plant as shown in Figure 1.

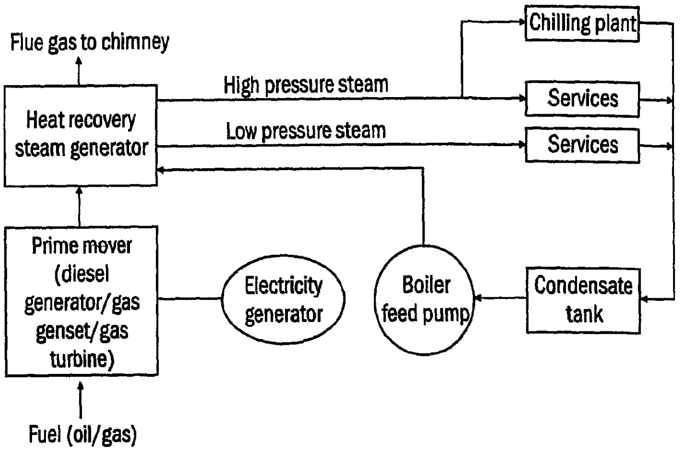

Figure 1 Typical cogeneration plant for hotel industry/commercial buildings

The prime movers, which can be used, are either diesel generators, gas gensets or gas turbines. Each of these has its capacity range and fuel preference.

Diesel-oil-fired generating sets normally used are for ratings from a few kilowatts to 2000 kW. However, larger rating sets are also available for generating power for medium and large industry. A typical output of electricity and steam for cogeneration plants is in the range of 1250–2000 kW (Table 1). It may be seen that only low temperature steam can be produced from such systems.

Natural gas-fired gensets are in the range of 500–3000 kW output. The engine efficiency is in range of 38%–42%, whereas the cogeneration efficiency is in the range of 65%–68%. A typical output of electricity and steam for the plants is given in Table 2. The steam produced is at 10 bar pressure, which is the normal requirement for such buildings.

Table 1 Cogeneration based on diesel generators

Electrical output (kW_e)	*Energy input* (kW_e)	*Electrical efficiency* (%)	*Generation from HR system*	*Steam pressure bar/hot water temp* (°C)	*Quantity steam/ hot water* (kg/h)	*Overall energy efficiency* (%)
1250	1269	41.2	Hot water	80	10000	66.5
1720	2194	32.6	Steam	15	892	53.8
2000	1921	38.5	Steam	5	208	44.8

Note Typical calculations made with the data supplied by HR (heat recovery) system manufacturers, while generators are opening (40%–100% load level) in North India.

Table 2 Cogeneration based on natural-gas-fired gensets

Electrical output (kW_e)	*Energy input* (kW_e)	*Electrical efficiency* (%)	*Steam from exhaust gas at 10 bar* (kg/h)	*Overall energy efficiency*[a] (%)
625	1615	38.68	420	65.8
836	2154	38.81	559	66.1
1048	2692	38.93	700	66.5
1461	3529	41.40	820	66.8
1944	4706	41.30	1094	67.2
2717	6481	41.42	1476	68.0

[a] Includes heat used from engine casing water circuit for heating BFW (boiler feed water) from 30 °C to 80 °C

Note A typical plant in North India (second row from the top) will give an output of 803 kW_e and 750 kg/h steam at 10 bar.

The gas turbines can use either diesel or natural gas. A capacity range of 500 kW to 250 MW is available today in the world, having open cycle efficiency from 20%–39%. However, in the capacity range adopted for small-scale industry/ commercial buildings, the efficiency is in the range of 20%–33%. Electricity and steam output from typical machines for this range is given in Table 3. It may be seen that steam output is high as compared to diesel generators or gas gensets for the same capacity ratings for electricity generation.

Table 3 Cogeneration based on natural gas-fired gas turbines (0.5–10 MW)

Electrical output (kW_e)[a]	*Energy input* (kW_e)	*Electrical efficiency* (%)	*Steam from exhaust at 10 bar* (kg/h)	*Overall efficiency* (%)
504	2 468	20.5	2 045	36.1
3051	11 559	26.4	9 864	36.2
3507	12 262	28.6	10 091	36.5
9194	27 800	33.1	19 227	46.6

[a] At ISO conditions - 15 °C, 60% relative humidity, sea level

Note A typical cogen plant in North India (first row from the top) will give an output of 460 kW_e and net steam of 1700 kg/h at 10 bar.

All the systems mentioned in Tables 1, 2, and 3 have the limitation of the maximum electric power it can generate depending upon the prime mover chosen. However, the steam capacity, if required, can be enhanced through supplementary firing in the heat recovery steam generators.

Selection of cogeneration systems for a hotel or commercial building

The various electric and thermal loads to be considered are listed below.

- *Electric load* lighting system, services, pumps, refrigeration, air cooling (with vapour compression system), kitchen appliances, etc.
- *Thermal load* (1) Steam for laundry, kitchen, health club and hot water (steam conditions for each application may be different); (2) steam for vapour absorption refrigeration machine/chillers

These loads have to be summed up to the generating source value by suitably incorporating the line losses with respect to the location and layout of different equipment. For example, all electrical losses in the switching and regulating systems have to be accounted for. Start-up current required for different motors has also to be considered. For steam lines, both pressure and thermal losses should be considered.

Based on the above parameters, electrical and steam load demand curves from 0 to 24 h are developed (Figure 2).

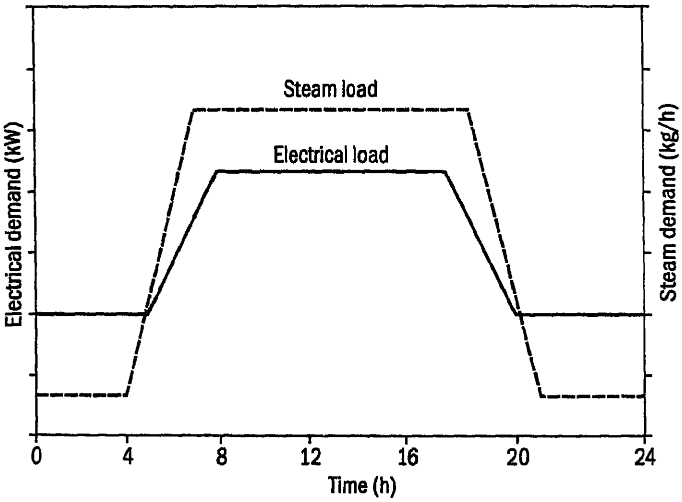

Figure 2 Electrical and steam load demand patterns

Now compare this demand (peak) for electricity and steam with the outputs given in Tables 1, 2, and 3. Wherever it matches suitably (with single or multiple units), that will be the best cogeneration system for your specific application. If the steam demand is a little more than the steam generation, it can be managed by having supplementary firing in the heat recovery steam generators.

Based on the above assessment and the part-load performance data for the prime mover obtained from the manufacturer, steam generation and the demand can be compared. It is generally found that for the kind of load curves given in Figure 2, part-load demands of steam can be met when the generator is running at part load.

For retrofit applications also, a suitable cogeneration system can be examined. The guiding principles remain the same as mentioned earlier.

Advantages of captive cogeneration units

The main advantages of captive cogeneration units are listed below.

- No dependence on the grid power supply for regular operation. Only for emergency or during plant maintenance, grid power may be required. This may be suitably negotiated with the electricity supply company.
- The cost of generation of power will be normally less than the purchase price of electricity; a typical cost of generation calculation is given in case studies I and II. The cost figures are only indicative. These have to be replaced with specific site data. The case studies only give the method of calculation.
- Vapour absorption refrigeration systems can be based on steam. Operating cost will be low.
- While designing the systems for retrofit applications, the existing connected load may be retained. Power corresponding to the minimum charges per month should be consumed. The balance load may be connected to the cogeneration plant.

Case study I: Cogeneration system based on gensets fired by natural gas

A typical peak-load requirement of a 300-room hotel is 1500 kW electric power. If natural gas supply is available/or can be made available through the pipeline network, the gas-fired gensets 2 × 836 kW, as given in (second row from the top), Table 2, can be chosen. Thus,

Electrical output = 2 × 836 = 1672 kW_e
Saturated steam output (with BFW [boiler feed water] temperature = 80 °C and heat recovery from casing) = 2 × 750 = 1500 kg/h at 10 bar

LCV (lower calorific value) of natural gas = 9220 kcal/SCM (standard cubic metre)
Natural gas price = Rs 14/SCM

Input energy (from
manufacturer's catalogue) = 2 × 1.8 = 3.6 million kcal/h
= 3 600 000/9220 = 390.5 SCM/h
PLF (plant load factor) = 0.85

Estimated cost of cogen plant = Rs 50 000 000
Cost of steam = Rs 1.25/kg
Interest on capital = 12%
O&M (overheads and
maintenance) cost (% of
capital per year) = 3%

The cost of generation of electricity can be calculated as follows.
Cost of capital per year = Rs 50 000 000 × 12/100
= Rs 6 000 000

Cost of O&M per year = Rs 50 000 000 × 3/100
= Rs 1 500 000

Cost of fuel = Rs 390.5 × 365 × 24 × 0.85 × 14
= Rs 40 707 282

Cost of steam = Rs 1500 × 365 × 24 × 0.85 × 1.25
= Rs 13 961 250

Electric power produced
per year = 1672 × 365 × 24 × 0.85
= Rs 12 449 712 kWh

Cost of electric power (Rs/kWh) =

$$= \frac{6\ 000\ 000 + 1\ 500\ 000 + 40\ 707\ 282 - 13\ 961\ 250}{12\ 449\ 712}$$

$$= \frac{34\ 246\ 032}{12\ 449\ 712}$$

$$= 2.75$$

Case study II: Cogeneration system based on gas turbine fired by natural gas

If the electric power requirement is less than the previous case, but the steam requirement is more (because of vapour absorption refrigeration system), a cogeneration system as given in (first row from the top), Table 3, can be chosen. The basic data for this system are as follows.

Electrical output = 2 × 504 = 1008 kW_e
Saturated steam output = 2 × 2045 = 4090 kg/h at 10 bar
Steam required for chilling
inlet air for gas turbine = 2 × 40 TR (tonnes of refrigeration)
= 2 × 40 × 5.3 kg/h steam
= 424 kg/h steam
(assumed that 5.3 kg steam required per TR)

Net steam available = 4090 – 424 = 3666 kg/h
Natural gas price = Rs 14/SCM
LCV of natural gas = 9220 kcal/SCM

Input energy (from manufacturer's catalogue) = 2 × 2.057 million kcal/h
= 446 SCM/h

PLF = 0.85
Cost of steam = Rs 1.25/kg
Estimated cost of cogen plant = Rs 60 000 000
Interest on capital = 12%
O&M cost (% of capital per year) = 3%

The cost of generation of electricity can be calculated as follows.

Cost of capital per year = Rs 60 000 000 × 12/100
= Rs 7 200 000

Cost of O&M per year = Rs 60 000 000 × 3/100
= Rs 1 800 000

Cost of fuel per year = Rs 446 × 14 × 365 × 24 × 0.85
= Rs 4 64 92 824

Cost of steam per year = Rs 3666 × 365 × 24 × 0.85 × 1.25
= Rs 34 121 295

Electric power produced per year = 1008 × 365 × 24 × 0.85 kWh
= 7 505 568 kWh

Cost of electric power (Rs/kWh)

$$= \frac{7\,200\,000 + 1\,800\,000 + 46\,492\,824 - 34\,121\,295}{7\,505\,568}$$

$$= \frac{21\,371\,529}{7\,505\,568}$$

$$= 2.85$$

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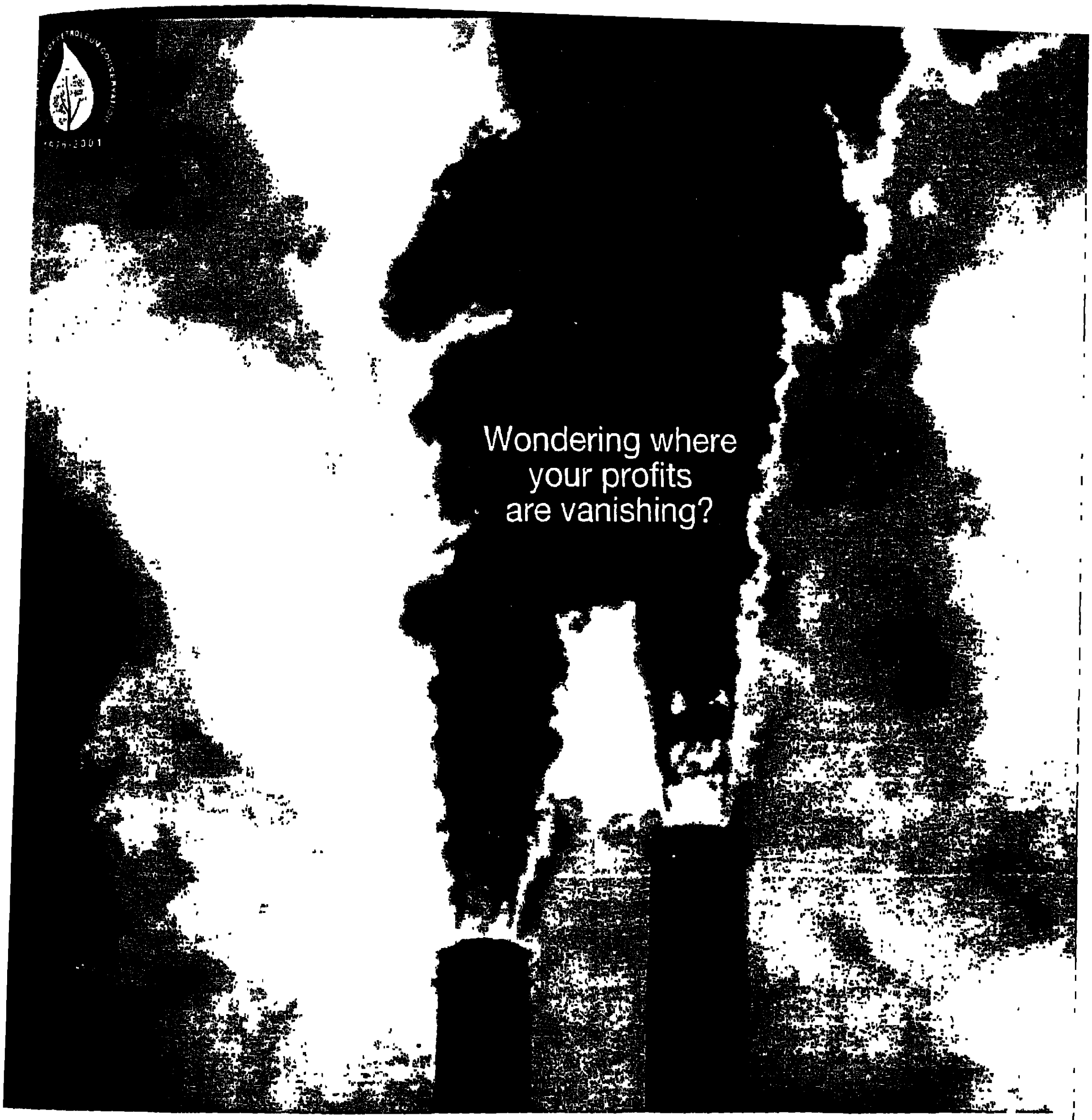

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e-mail : jaimahal.jaipur@tajhotels.com

The Rambagh Palace
Bhawani Singh Road, Jaipur - 302 005
Tel No. (91-141) 38 1919
e-mail : rambagh.jaipur@tajhotels.com

Annexure 1

Instruments

Name of the instrument	*Make*
Gas analysers	
Spectro 1600	MRU, Germany
MRU flue gas analyser	MRU, Germany
Land duo flue gas analyser	LAND, UK
Temperature measurement	
Sling thermometer	Jaspin, India
Heat spy	Wahl, USA
Non-contact temperature indicator	Airflow, UK
Pyrometer IE-7	Impac, Germany
Temperature indicator k type	Audiotronics, India
Temperature indicator TP 200	Eutech, USA
Data logger	Data electronics USA
Temperature indicator TTX 483	Eutech, USA
Air flow measurement	
Hot wire anemometer TA-6000	Air flow UK
Vane anemometer	Japsin, India
TA5 thermal anemometer	Vayubodhan, India
Distance measurement	
Measuring tape	Freeman, India
Electronic distance meter	Polaroid, USA
Water quality measurement	
Waterproof Ph TESTR 2	Eutech, USA
Waterproof TDScan WP 2	Eutech, USA
Electrical measurement	
ITT clampon multimeter MX-1200	ITT, USA
Fluke single phase power meter	Fluke USA
AR5 power and harmonic analyser	Circutor, Spain
Illumination level measurement	
Lux meter LX-101	Digital, Taiwan
Analog lux meter	Kyoritsu, Japan
Speed measurement	
Digital tachometer DT-2001B	Digital Promoters, India
Flow measurement	
Ultrasonic flowmeter DDF-3088	Polysonics, USA
Transit time flowmeter	Micronics, UK
Pressure measurement	
PDM - 210 (air)	Neotronics, UK
Water pressure measurement	Comark, UK

Continued

Name of the instrument	*Make*
Steam trap measurement	
Ultra troubleshooter	Ue Systems, USA
Leak check	ICC Federated, USA
TLV trapman TM-2	TLV Trapman, USA
Time measurement	
Digital stop watch	Jaspin, India

Annexure 2

Equipment suppliers / laboratories / government agencies

Absorption chillers

Blue Star Ltd
Kasturi Building
J Tata Road
Mumbai – 400 063

Thermax Ltd
Absorption Cooling Division
Chinchwad
Pune – 411 019

Voltas Ltd
19, J N Heredia Marg
Ballard Estate
Mumbai – 400 038

Air curtains

Air Devices Corporation
12/A, Bombay Shopping Centre
Race Course Road
Baroda – 390 005

Jet India
218, Allied Industrial Estate
Off MMC Road Mahim (W)
Mumbai – 400 016

Air flow meters

Air Devices Corporation
12/A, Bombay Shopping Centre
R C Dutta Road
Baroda – 390 005

J N Marshall Systems and Services
P B No. 1, Mumbai–Pune Road
Kasarwadi, Pune – 411 018

Toshniwal Sensors Pvt. Ltd
E/19-20, Makhupura Industrial Area
Ajmer – 305 002

Techmark Engineers and Consultants
K-1/28, Ground floor
Chittaranjan Park
New Delhi – 110 019

Automatic drain valves for compressed air systems

Associated Pneumatics
1, PSG Estate Colony
Peellamedu, Coimbatore – 641 004

Prasad Machine Works
8/85, Nehru Nagar
Post Kalaputti
Coimbatore – 641 035

Cell-type air washers (humidification)

Nachmo Polyplast Ltd
11, Ashima House
Kavi Nanalal Marg
Ahmedabad – 380 006

Roots Cooling Systems Pvt. Ltd
A-78, Sector 4
Noida – 201 301

Clip-on type kW and PF meters, electronic meters, and computerized energy management systems

Alacrity Electronics Ltd
15, Thirumalai Road
T Nagar, Chennai - 600 017

Asea Brown Boveri Ltd
22-A, Shah Industrial Estate
Off Veera Desai Road
Andheri (W), Mumbai

Enercon Systems Pvt. Ltd
23, KHB Light Industries Area
PO Box No. 6418, Yelahanka
Bangalore - 560 064

Larsen & Toubro Ltd
L & T Chennai House
10, Club House Road
Anna Salai, P B No. 5247
Chennai - 600 002

Meco Instruments Pvt. Ltd
301, Bharat Industrial Estate
P O Box 6388, T J Road Sewree (W)
Mumbai - 400 015

Merlinhawk Associates Pvt. Ltd
134 / 1, Brigade Road
Bangalore - 560 025

Microtek Instruments
40-A, 1st Main road
CIT Nagar, Chennai - 600 035

Motwane Private Ltd
127, Mahatma Gabdhi Road
P O Box No. 1312, Fort
Mumbai - 400 023

Secure Meters Ltd
P B No. 30
Pratap Nagar Industrial Area
Udaipur - 313 003

Siemens Ltd (MDA Department)
Electric Mansion
1986, Appasaheb Marathe Marg
P B No. 1911, Prabhadevi
Mumbai - 400 025

Yokogawa Blue Star Ltd
203, Somdutt Chambers II
9 Bhikaji Cama Place
New Delhi - 110 066

Compressed air control systems

Godrej & Boyce Mfg Co. Ltd
E&E Services, Pirojshanagar, Vikhroli
Mumbai - 400 079

Compressed air moisture separators

Controquip Pvt. Ltd
2nd Floor, Visharad Complex
Near High Court, Ashram Road
Ahmedabad - 380 009

Flat belts

Pavitra Fabricators Pvt. Ltd
62, Usha, 161, SV Road
Vile Parle (W), Mumbai - 400 056

Polygrip International
506, Silver Oak's Com. Complex
Mahalaxmi Char Rasta, Paldi
Ahmedabad - 380 007

Finned coils heat exchangers

Coil Co. Pvt. Ltd
A-21/24, Naraina Industrial Area
Phase - II
New Delhi - 110 028

Thermotech Industries
13, Vishal Industrial Estate
Village Road, Bhandup
Mumbai - 400 078

High efficiency fans

Asea Brown Boveri Ltd
Guru Nanak Foundation Building
15–16, Qutub Institutional Area
New Delhi – 110 067

Thermax Ltd
Environment Division
Chinchwad, Pune – 411 019

TLT Engineering India Pvt. Ltd
64, GIDC Industrial Estate
Vatva, Ahmedabad

Universal Air Systems Pvt. Ltd
1, Jalaram Commercial Centre
Saki Vihar Road, Saki Naka
Mumbai – 400 072

High efficiency FRP cooling tower fans

Cooling Tower Corporation
40, GIDC, Makarpura, Baroda – 390 010

Melfrand Engineers Pvt. Ltd
Govind Nagar, Off N S S Road
Home Guard Centre, Ghatkopar (W)
Mumbai – 400 084

Paharpur Cooling Towers Ltd
Paharpur house, 8 / 1/ B
Diamond Harbour Road
Kolkata – 700 027

Heat exchanger non-chemical descaling

Inteco Colloids Pvt. Ltd
606, Eros Place
56, Nehru Place, New Delhi – 110 019

Scalemaster Adlam Pvt. Ltd
11, Vasavi Nagar Colony
Secunderabad – 500 003
(Electrostatic Water Descaler)
(Mechanical Colloid Formation)

Infrared heating system

Litel Infrared System Pvt. Ltd
J-284, MIDC, Bhosari
Pune – 411 026

Lighting (lamps, luminaires, and electronic ballast)

Bajaj Electricals Ltd
15/17, Sant Savta Marg
Reay Road, Mumbai – 400 010

Crompton Greaves Ltd
(Lighting Division)
Dr E Morses Road
Worli, Mumbai – 400 018

GE Lighting
Maker Chambers III, 1st Floor
Jamanalal Bajaj Road
New Delhi – 110 015

Osram India Pvt. Ltd
1/95, Market Road
Bhai Veer Singh Lane
New Delhi – 110 001

Philips India Ltd
M 38-1 Middle Circle
Connaught Place
New Delhi – 110 001

Surya Roshni Ltd
Padma Tower – 1, Rajendra Place
New Delhi – 110 008

Wipro Lighting
Tulsi Chambers, Opp. St Francis School
Jalna Road, Aurangabad – 431 001

Lux meter (for illuminance levels)

Digital Promoters (India) Pvt. Ltd
505, Vishal Bhawan
95, Nehru Place, New Delhi – 110 019

Conin Prakriti Instrumentation
16, Rajendranagar Industrial Estate
Post Mohan Nagar
Ghaziabad – 201 007

Research Instrumentation
A-10, Naraina Industrial Area
Phase – I, New Delhi – 110 028

Occupancy sensors/controls/ building automation

Philips India Ltd
M 38-1 Middle Circle
Connaught Place, New Delhi – 110 001

Pyrotech Marketing and Projects Pvt. Ltd
412, Shakuntala Apartments
59, Nehru Place, New Delhi – 110 019

Shivalika Proenergetics Ltd
A-116, Madhuvan, Vikas Marg
New Delhi – 110 092

Tata Honeywell Ltd
917, International Trade Tower
Nehru Place, New Delhi – 110 019

Power factor controllers/indicators

Neutronics Manufacturing Co.
12-A, Maril Maroshi Road
Opp. Old State Bank, Andheri (E)
Mumbai – 400 059

Process Technique Electronics Pvt. Ltd
324 'Sandhilya', P B No. 1776
Vimanpura Post, Bangalore – 560 017

Pressure guages / manometers / pitot tubes

Instrument Research Associates (P) Ltd
P B No. 2304, 228, Magadi Road
Bangalore – 560 023

J N Marshall Systems and Services
P B No. 1, Bombay Pune Road
Kasarwadi, Pune – 411 018

Manometer (India) Pvt. Ltd
Manu Mansion
16, Shahid Bhagat Singh Road
Mumbai – 400 022

Techmark Engineers and Consultants
K-1/28, Ground Floor, Chittaranjan Park
New Delhi – 110 019

Vahyubodhan Upkaran Pvt. Ltd
A-292 / 1, Okhla Industrial Area
Phase – I, New Delhi – 110 020

Plate heat exchangers

Alfa-Laval (India) Ltd
Dapodi, Pune – 411 012

Blacke-Durr Kaveri (P) Ltd
Shanthalu Plazza
146, 8th Main Road, Malleswaram
Bangalore – 560 003

Indswep Energy Systems (P) Ltd
C-6/12, Salunke Vihar
Kondhwa, Pune – 411 048

Relative humidity meters

Aimil Ltd
Naimex House, A-8
Mohan Co-operative Industrial Estate
Mathura Road, New Delhi – 110 044

Digital Promoters (India) Pvt. Ltd
505, Vishal Bhawan
95, Nehru Place, New Delhi – 110 019

Instrument Research Associates
P B No. 2304, 228, Magadi Road
Bangalore

OPTEL
104, Brahmanwadi
K A Subramanian Road
Matunga, Mumbai – 400 019

Recuperator for waste heat recovery

Encon Thermal Engineers (P) Ltd
308, Race Course Towers
Race Course Circle
Baroda – 390 015

Sun control films

Garware Plastics and Polyster Ltd
Western Express Highway
Vile Parle
Mumbai – 400 057

Soft starters with energy saving feature

Allen-Bradley India Ltd
C-11, Industrial Area
Site-4 Sahibabad
Ghaziabad – 201 010

Bharat Bijlee Ltd
Industrial Electronics Division
501-502, Swastik Chambers
Chembur, Mumbai – 400 023

Crompton Greaves Ltd
Industrial Electronic Division
71/72, MIDC Industrial Area
Satpur, Nashik – 422 007

Jeltron Systems (India) Pvt. Ltd
Bag No. 49, 6-3-99/2
Vaman Nayak Lane, Umanagar Colony
Begumpet, Hyderabad – 500 018

Siemens Limited (MDA Dept)
Electric Mansion
1986, Appasaheb Marathe Marg
P B No. 1911, Prabhadevi
Mumbai – 400 025

Temperature monitors/controllers/ sensors

Aimil Ltd
Naimex House, A-8
Mohan Cooperative Industrial Estate
Mathura Road
New Delhi – 110 044

Digital Promoters (India) Pvt. Ltd
505, Vishal Bhawan
95, Nehru Place
New Delhi – 110 019

Instronix India Pvt. Ltd
3E / 12, Jhandewalan Extension
New Delhi – 110 055

Masibus Process Instruments (P) Ltd
B/30, GIDC Electronic Zone
Gandhinagar – 382 044

Techmark Engineers and Consultants
K-1/28, Ground Floor, Chittaranjan Park
New Delhi – 110 019

Toshniwal Sensors Pvt. Ltd
E/19-20, Makhupura Industrial Area
Ajmer – 305 002

Time/control switches (timers)

Escol Electromech Pvt. Ltd
Kripalani Estate
Saki Vihar Road
Mumbai – 400 072

Indo Asian Marketing Ltd
A-59, Okhla Industrial Area
Phase II
New Delhi – 110 020

Jayshree Enterprises
(Marketing Division)
101, Prabodhan Apartment
64/9, Erandwane
Pune – 411 004

Larsen & Toubro Ltd
P O Box No. 8901, Powai
Mumbai – 400 072

Ultrasonic liquid flow meters

Glentec Instrumentation
16, Neelkanth Complex
Chembur Govandi Road
Chembur
Mumbai – 400 071

Josts Engineering Company Ltd
F-18, Ground Floor, Lajpat Nagar III
New Delhi – 110 024

J N Marshall Systems and Services
P O Box No. 1
Mumbai–Pune Road, Kasarwadi
Pune – 411 018

Techmark Engineers and Consultants
K-1/28, Ground floor, Chittaranjan Park
New Delhi – 110 019

Toshniwal Bros. (Bombay) Pvt. Ltd
Ketan Building
46, Shahid Bhagat Singh Road
Mumbai – 400 023

Variable speed drives

Allen Bradley India Ltd
A-5, Second Floor, Kailash Colony
New Delhi – 110 048

Asea Brown Boveri Ltd
Guru Nanak Foundation Building
15–16, Qutub Institutional Area
New Delhi – 110 067

Crompton Greaves Ltd
Industrial Electronic Division
71/72, MIDC industrial Area
Satpur, Nashik – 422 007

Jeltron Systems (India) Pvt. Ltd
Bag No. 49, 6-3-99/2
Vaman Nayak Lane, Umanagar Colony
Begumpet, Hyderabad – 500 018

Kirloskar Electric Company Ltd
Unit IV, Belawadi Industrial Area
Mysore – 570 005

L&T Static Control
(Electronic Contols Division)
Larsen & Toubro Ltd
P O Box No. 8901, Powai
Mumbai – 400 072

Siemens Ltd (MDA Dept)
Electric Mansion
1986, Appasaheb Marathe Marg
P B No. 1911, Prabhadevi
Mumbai – 400 025

The General Electric Company of India Ltd
Electronics Unit, DA 21, Salt Lake City
Kolkata – 700 064

Laboratories for calibration of instruments

Electrical Research and Development
Association
P O Box No. 760 GIDC
Industrial Estate Makarpura
Baroda – 390 010

Fluid Control Research Institute
Kanjikode West P O
Palakkad – 678 623, Kerala

Government agencies supporting energy conservation activities

Energy Management Centre
118, Ashirwad Complex
D-1, Green Park
New Delhi – 110 016

Indian Renewable Energy Development Agency Ltd
India Habitat Centre Complex, Core-4A,
East Court, First Floor, Lodhi Road
New Delhi – 110 003

Industrial Credit and Investment Corporation of India Ltd
C-23, G Block, Bandra–Kulra Complex,
Bandra (East), Mumbai – 400 051

Industrial Development Bank of India
IDBI Tower, Cuffe Parade
Mumbai – 400 005

Maharashtra Energy Development Agency
MHADA Commercial Complex
S No. 191-A, Phase – I
Opp. Tridal Nagar, Yerawada
Pune – 411 006

Madhya Pradesh Urja Vikas Nigam
B-Block, GTB Complex
T T Nagar, Bhopal – 462 003

Ministry of Non-conventional Energy Sources
Block No. 14, CGO Complex
Lodi Road, New Delhi – 110 003

Petroleum Conservation Research Association
New Delhi House
27 Barakhamba Road
New Delhi – 110 001

Conversion factors

- 1 kcal = 3.9685 Btu
- 1 kWh = 3413 Btu
- 1 kWh = 860 kcal = 3.6 MJ
- 1 Btu = 1.055 kJ
- 1 Calorie = 4.186 joules
- 1 Joule = 1 watt-second
- 1 HP = 746 watts
- 1 kg = 2.2 lb (pounds)
- 1 metre = 3.28 feet
- 1 inch = 2.54 cm
- 1 kg/cm^2 = 14.22 psi
- 1 atmosphere = 1.0332 kg/cm^2
- 1 kg/cm^2 = 28.99 inches of Hg @ 0 °C
- 1 kg/cm^2 = 32.79 inches of H_2O @ 4 °C
- 1 kg/cm^2 = 10 inches of H_2O @ 4 °C
- 1 kg/cm^2 = 9.807×10^4 pascals
- 1 tonne of refrigeration = 3023 kcal/hour
- 1 tonne of refrigeration = 12 000 kcal/hour
- 1 tonne of refrigeration = 3.51 kW
- 1 US gallon = 3.785 litres
- 1 imperial gallon = 4.546 litres
- °K = 273° + °C

SI prefixes

Multiplication factors	*Prefix*
10^{18}	Exa
10^{15}	Peta
10^{12}	Tera
10^{9}	Giga
10^{6}	Mega
10^{3}	Kilo
10^{2}	Hecta
10^{1}	Deca
10^{-1}	Deci
10^{-2}	Centi
10^{-3}	Milli
10^{-6}	Micro
10^{-9}	Nano
10^{-12}	Pico
10^{-15}	Femto
10^{-18}	Atto

Calorific values for various fuels (1 tonne of oil equivalent = 10.2 million kcal)

- 1 million tonnes of coal = 0.49 MTOE (million tonnes of oil equivalent)
- 1 mm of natural gas = 856.9 TOE
- 12 000 GWh of hydro energy = 1 MTOE
- 12 000 GWh of nuclear energy = 1 MTOE
- 1 MT of high-speed diesel = 1 MTOE
- 1 MT of light diesel oil = 1 MTOE
- 1 MT of fuel oils = 1 MTOE

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Mather and Platt (India) Ltd	p. 90
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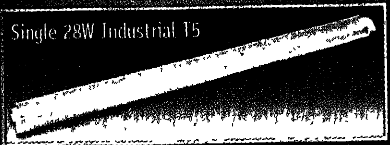
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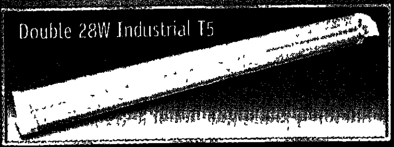
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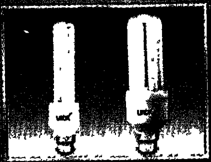

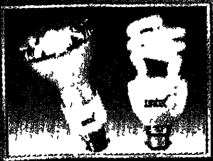

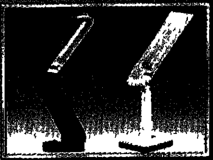

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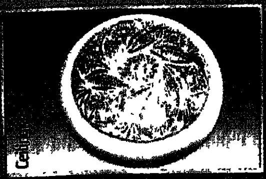

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